COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

APR 8 1944

March, 1944

Photo by H. R. Towse

Power plant and recovery room of the Chesapeake Corporation pulp and paper mill at West Point, Va. Discharge from stack shows characteristic water vapor resulting from burning black liquor as fuel

Addition to Capacity and Modernization of the MARYSVILLE POWER HOUSE—IV

Control Equipment for Single-Retort Underfeed Stokers

Design Data for Overfire Jets

WHEN PEAK LOAD RATING BECAME CONTINUOUS DEMAND

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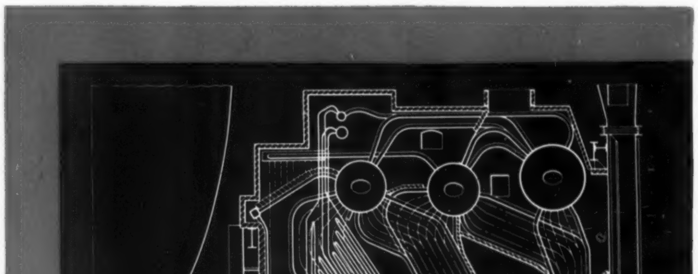

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DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME FIFTEEN NUMBER NINE

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EDITORIAL

Disposal of Surplus Property

The U. S. Chamber of Commerce has released a digest of a special committee report dealing with the disposition of surplus war plants, stocks and facilities, which, in general, is in harmony with the Baruch and Senate Post-War Committee reports.

Based on the broad premise that the Government should withdraw from business operations in favor of private ownership at the close of hostilities, the committee cautions against speedy liquidation such as might interfere with conversion of our economy to a normal peacetime basis, and suggests instead that a gradual and orderly procedure be employed in the disposal of properties and surpluses. It believes this can be accomplished best by legislation setting up an independent commission to administer the demobilization program and by the establishment of basic principles and policies to guide the commission in the conduct of its operations.

Other points advanced in the report are: (1) that an early decision be made as to plants and equipment that should be retained for military purposes and the remainder earmarked for orderly disposal; (2) that disposals be handled in such a manner as to encourage competitive efficiency on the part of private industry without giving preferential advantage to any locality or subsidizing any particular enterprise; (3) that properties be sold rather than leased; (4) that disposal of machinery and equipment be made available in small lots or individual units, as demand arises; and (5) that in so far as practicable, surpluses be redistributed by the industries which produced them.

These recommendations appear both reasonable and logical, and as least likely to disturb an orderly return to peacetime economy. However, when dealing with some classes of equipment, they may be applicable only in part, as for instance, certain power plant equipment. It will be recalled that, in the process of outfitting war plants, unavailability of new units made it necessary to dismantle many boilers, turbines and other power equipment in non-essential use or in reserve and to re-erect them in war plants. In most instances, due to the labor involved, costs exceeded those of new equipment. At that time necessity outweighed expense, but this would not hold for reconversion. Hence, it would not be economical to salvage much of this older equipment, especially when it is considered that many of these units had already achieved a considerable degree of depreciation and obsolescence at the time of their installation in war plants. New equipment, of course, falls in a different category.

While the war is far from won, it is nevertheless as important to prepare for peace as it was to prepare for war. With the Chamber of Commerce, the Baruch Committee and Congress apparently in accord on the main objectives, there remains to be settled the question of administrative authority and then the vast amount of detail in organization and execution of the plan that is adopted. The thought that has been given to this problem from competent sources promises to avoid many of the pitfalls that occurred at the end of the last war.

Burning Mixtures of Bituminous Coal and Anthracite Fines

Increased production of anthracite, resulting partly from the seven-day work week during February in an effort to meet domestic demands, has added greatly to the surplus of fine sizes such as No. 3 buckwheat and smaller. At some of the mines these surpluses, which are unfit for domestic use, threaten to clog existing facilities unless a way can be found to move them more readily. Hence the Solid Fuels Administration has set about to find additional outlets for this growing accumulation which now amounts to over a million tons of freshly mined coal. It is proposing that industrial power plants, equipped with underfeed stokers, burn a mixture of bituminous coal and anthracite fines up to ten or twenty per cent of the latter. This, it contends, would help alleviate the present situation as to the low-volatile bituminous coal supply.

Such small sizes of anthracite have long been burned extensively on traveling-grate stokers; but, despite the large number of such installations, they apparently are not sufficient to absorb the present production of this fuel. The idea of burning mixtures of bituminous coal and anthracite fines on underfeed stokers is not new; it has been practiced for a number of years in a few large plants equipped with multiple-retort underfeed stokers and in some smaller plants having single-retort stokers.

A major difficulty in firing such mixtures lies in avoiding segregation in the bunker and in feeding, so as to assure that the fuel will be well mixed when reaching the stoker. If segregation occurs in the fuel bed, the air pressure suitable for burning the bituminous coal will be too high for the anthracite, so that the latter will be blown over into the dumps and holes will occur in the fuel bed. Furthermore, while the bituminous coal will coke at the front of the stoker, the anthracite will not, and there is the likelihood of the latter passing through the overfeed section at the rear (in the case of multiple-retort stokers) without being completely consumed.

Furnaces and grates are differently proportioned for burning each of the two fuels; but, assuming that the amount of anthracite is held down to ten or fifteen per cent, it may be possible, as an emergency measure, to obtain fairly satisfactory results under proper control at some sacrifice in efficiency. However, it should be pointed out that the average industrial plant lacks facilities for assuring a thorough and constant mixture, and for exercising proper control over the proportions.

Control Equipment for Single-Retort Underfeed Stokers

By H. G. MEISSNER
Combustion Engineering Company, Inc.

Based on a review of the functioning of a single-retort underfeed stoker, the several types of control are described and compared, and practical suggestions as to installation and operation are given.

THE single-retort underfeed stoker is one of the oldest and most widely used types of mechanical stoker. It was invented some time before the middle of the nineteenth century, one of the earliest known designs having been patented by John Juckes in England in 1838. It is interesting that all four of the most popular types of mechanical stokers in use today—traveling grate, overfeed, spreader and underfeed—were developed and patented in England in the relatively brief period from 1822 to 1841.

Today, the single-retort underfeed stoker is in first place, both in number of existing installations, and in current sales—a remarkable record in this era of rapidly changing ideas and revolutionary improvements in practically every art.

Its popularity is well deserved, and in spite of increasing competition from other methods of firing boilers, such as oil and gas burners, pulverized coal and traveling grates and spreader stokers, it seems likely to remain in the foreground of the coal-burning picture as long as there is suitable coal to burn.

Constant improvement has been made in the design and construction of the underfeed stoker, since its inception, although the fundamental design as shown in the early models was sound, and has been retained practically unchanged. One of the greatest improvements in mechanical firing, made largely in the past quarter century, has been the development of automatic controls, to supplement and supersede, in so far as practicable, the manual adjustments formerly necessary. The use of these controls has progressed to the point where, in certain applications, a stoker will operate for days at a time, without manual attention, feeding and burning the coal, removing the ash, and maintaining desired pressures and temperatures.

A thorough knowledge of the various types of controls is therefore desirable, and in fact essential, to assure the proper selection in the design of a new plant, and to secure the optimum performance from existing equipment. The following discussion is designed to assist the designer, purchaser, and operator, to get the maximum results from control equipment now commercially available.

Underfeed stokers are used in probably the widest range of applications of any fuel-burning equipment from steam, hot-water and hot-air domestic heaters, to boiler furnaces in large industrial and utility plants. This discussion will be limited to applications larger than domestic units which present a control problem peculiar to the domestic heating field.

The controls for these stokers include start and stop types actuated by pressure or temperature, or sometimes a combination of pressure and temperature; high- and low-, or two-speed, controls also actuated by thermostats, aquastats, or pressurestats; high- and low-water, overpressure, hold fire and banking controls; positioning regulators, usually energized from steam pressure; and complete combustion controls in which the air flow is adjusted to maintain a predetermined fuel-air ratio, and the furnace draft is automatically regulated.

These controls may be powered by electric motors acting directly on the dampers and levers; by mechanical or mercury switches which make and break the circuits; or by compressed air, oil or water under pressure in the power pistons attached to the parts to be regulated. It will be seen, therefore, that the controls involved include a wide range of mechanical and electrical motions, although each comprises a simple element.

An understanding of the function of the various parts comprising an underfeed stoker and a knowledge of what happens in the fuel bed are useful in analyzing the function of automatic controls, and in establishing their limitations. There are certain things which can only be done by human hands, and the controls are designed to assist rather than to supersede the fireman.

A typical stoker is shown in Fig. 1. The coal is fed from the hopper to the front of the retort, by means of a feeder block. The length of the stroke of this feeder block is constant, but the number of strokes per minute is automatically controlled to suit the required coal feed.

Functioning of Single-Retort Stoker

As the coal enters the retort, it is picked up by the auxiliary pushers in the bottom of the retort and moved along toward the rear, at the same time being pushed upward and flowing over the top of the fire bars. The stroke of the auxiliary pushers corresponds with that of the feeder block, but their position and their height above the sliding bottom are manually adjustable, and must be changed to suit the fuel being fired, so that a uniform fuel bed from front to rear will be secured.

The coal is gradually heated as it moves up in the retort, and the gases begin to distill off as the coal ap-

proaches the fire bars. Air from the main air chamber is admitted to the fuel bed at this point, and ignition and active combustion commence. It is important that the proper amount of air be supplied here, so that the volatile constituents in the coal, which cause coking and caking, will be driven off and consumed. The actively burning fuel is then carried toward the dump grates by gravity and the reciprocating motion of the moving bars. These bars move in synchronism with the feeder block and pushers, but their stroke is manually adjustable, to suit the particular coal being burned. When the length of stroke is excessive, the fuel is moved toward the dump grates too quickly to be entirely consumed, while insufficient stroke causes thickening of the fuel bed over the retort, and a thin fire toward the side dumping grates. While the automatic controls can regulate the number of strokes per minute, the operator must make any necessary adjustments in length of stroke to maintain the correct fuel-bed contour.

Provision for relative adjustment of air flow to the retort end, and to the auxiliary air chamber, permits the fireman to change this proportion so that a well-burned-out refuse is deposited on the dumps. The automatic controls will supply the required total air to burn the fuel supplied, but the proportioning of this air through the fuel bed must be accomplished by the operator, based on observation of the fire.

During the dumping period, the fire door and ashpit doors must be open, and the automatic control will compensate for the resulting drop in furnace draft, by increasing the uptake damper opening. Any loss in steam pressure will cause the "master" regulator to increase the coal feed and forced draft to compensate for this condition. It is usually necessary for the fireman to reduce manually the coal feed while burning down and dumping the fire to avoid overfeeding during this period. There is generally sufficient fuel on the grate to take care of the load, and the fuel bed can be built up again after the fires have been cleaned, by manual adjustment of the fuel feed.

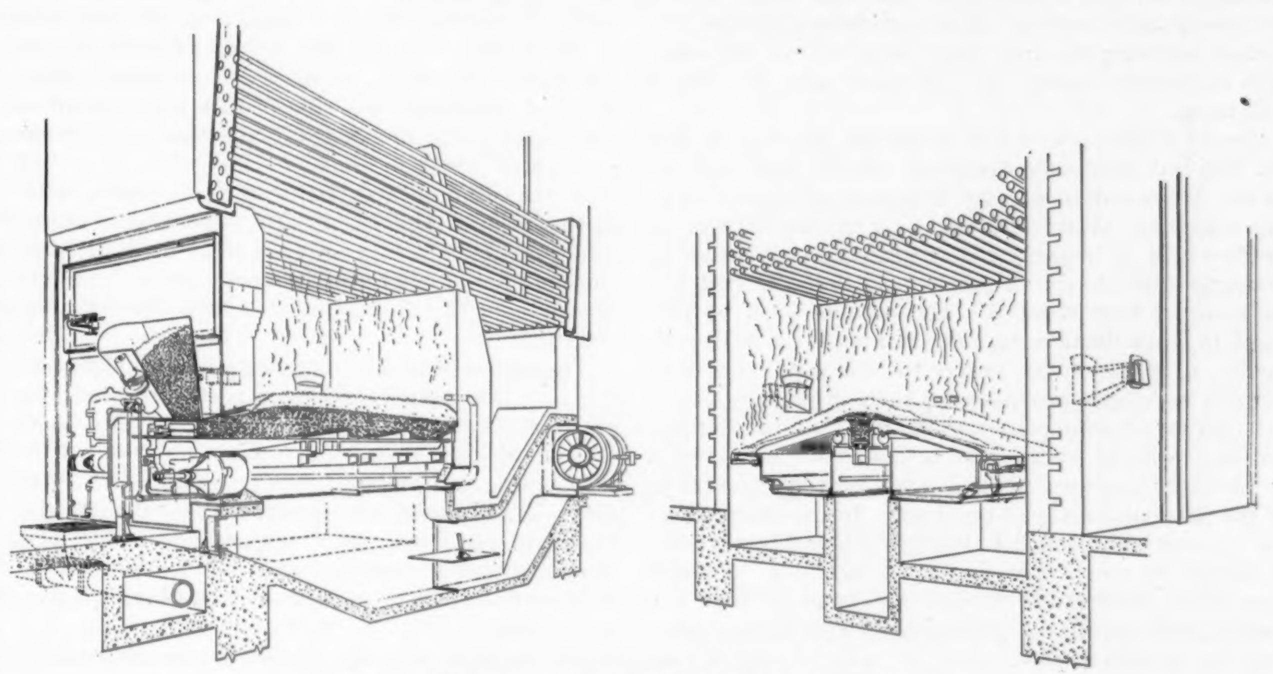


Fig. 1.—Typical single-retort underfeed stoker setting
Perspective views showing normal fuel bed contours, and mechanism used to feed and distribute coal and to regulate combustion air supply

Start-and-Stop Controls

The "Start-and-Stop" controls are widely used on the smaller underfeed stoker units because they are simple, reliable and flexible. As shown in Figs. 2a and 2b they consist of a mercury contact switch, which may be actuated by changes in pressure or temperature, wired through a transfer switch and starter box to the motor driving the stoker and fan. As the pressure or temperature rises to the upper limit desired, the circuit is broken, which stops the motor; and when the lower limit has been reached, the circuit is closed, restarting the motor. In this system it is necessary that fuel feed and forced draft be set at a point somewhat above that required to carry the maximum load, so that when the lower limit switch cuts in, the heat input will be such as to bring the pressure or temperature back to normal without undue delay.

The heat input of the stoker is therefore never exactly in step with the load on the system, that is, it is somewhat too high when the control is "on" and too low when the control is "off." However, the fuel bed is of sufficient mass to exert a flywheel effect, which carries the load along for some time after the stoker has stopped, and which prevents too sudden an increase when the control goes back to the "on" position.

High- and Low-Speed Control

The "high-and-low," or "two-speed," control (Fig. 3) as sometimes used on certain types of stoker units, maintains an active fire at all times. Instead of stopping the

stoker when the pressure rises to the upper limit, the stoker is slowed down to an adjustable lower rate of feed, which is somewhat below the minimum requirement. The coal burning rate can therefore be kept more nearly in step with the load, so that more efficient operation is obtained. This high-and-low operation is accomplished by mechanical adjustment of the variable-speed transmission on the stoker drive.

Positioning Controls

"Positioning-type" controls, as illustrated in Figs. 4a and 4b, adjust the fuel and air to conform closely with the boiler output, by means of a "master" controller which is normally actuated by steam pressure, and is mechanically connected to the fuel and air supply levers. A furnace-draft controller adjusts the uptake draft to maintain proper furnace conditions. For any given load the "master" takes a definite position, such that the fuel and air supply are just sufficient to match the steam output. By careful initial setting and occasional readjustment to correct for changes in fuel, this type of control will maintain reasonably correct fuel-air ratios over the normal load range.

Should a thin spot or hole in the fire develop, so that the fuel-bed resistance decreases, the air flow will increase, as the output of a fan is in inverse proportion to the resistance at its discharge. A similar change in air flow will be experienced should the coal sizing or characteristics change materially, so that the fuel bed resistance is thus reduced. The fireman must be prepared to make the necessary corrections in the coal or air supply, to maintain the proper fuel-air ratio, and most controls have easy provision for such adjustments.

Power for actuating the individual coal and air supply, and draft-control levers, may be supplied by air, water or oil under pressure, or by electric motors, depending on the particular make of regulator. In the illustrations the "master" is assumed to have an integral power unit. It should be noted that in the "positioning" controls there is no connection between the "master" and the furnace-draft regulators, as compared with the metering controls, described later.

Positioning-type controls are generally simple, reliable, easily understood and adjusted. They are probably the most satisfactory type for the smaller plants or where skilled operators are not available.

Metering or Proportioning Control

In the "metering or proportioning" control the gas flow through the boiler is metered or measured by balancing the pressure drop or draft loss through an orifice such as the boiler passes, air heater, or wind-box damper, across a spring or weight-loaded diaphragm, bellows, or float. This pressure drop varies as the square of the gas or air flow, and serves as an accurate measure of the rate of flow. For any given position of the "master," the rate of coal feed is fixed, as is the spring loading of the air-flow controller. Movement of the diaphragm and attached linkage causes the power unit to regulate the forced-draft supply in accordance with previously made calibrations.

The metering type control changes either the fuel feed or air supply to suit the load swings, and then proportions the fuel and air to a fixed ratio for each position of the master. In this way normal changes in fuel-bed thickness, coal sizing or other factors which affect the fuel-bed resistance will not disturb the fuel-air ratio. For example, the fuel-bed thickens somewhat because of prolonged operation since the last cleaning period, so that its resistance increases, with a consequent reduction in the air flow into the furnace. The air-flow controller notes this reduction in air flow and corrects this condition by increasing the fan damper opening, until the air flow returns to the amount for which the diaphragm is adjusted.

The preferred type of metering control is illustrated in Fig. 5a. The steam pressure actuates, through a spring or weight-loaded diaphragm or bellows, the "master" which positions the fuel-feed mechanism to suit the boiler load and steam pressure requirements. The "master" also resets the air-flow controller to move the forced-draft fan damper to supply the air requirements. The furnace-draft controller readjusts the uptake damper or the induced-draft fan, so that draft conditions are restored to normal.

For example, a drop in steam pressure causes the "master" to change position to increase the rate of coal feed, and increase the tension on the air-flow controller

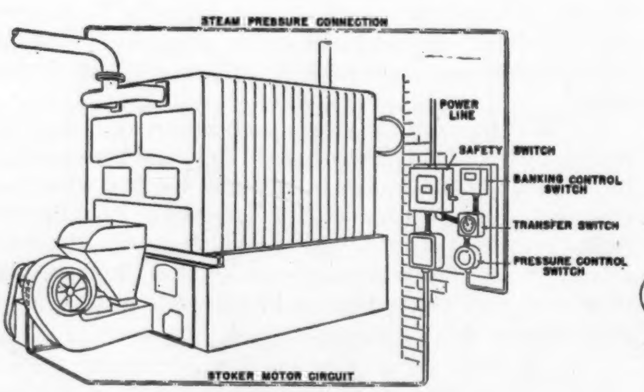


Fig. 2a—"Start-and-stop" control applied to low-ram stoker

Note simple compact panel board, with provision for regulation of stoker by steam pressure, thermostat, aquastat, bank control, etc., through transfer switch. Modulating or step-action regulators may also be applied.

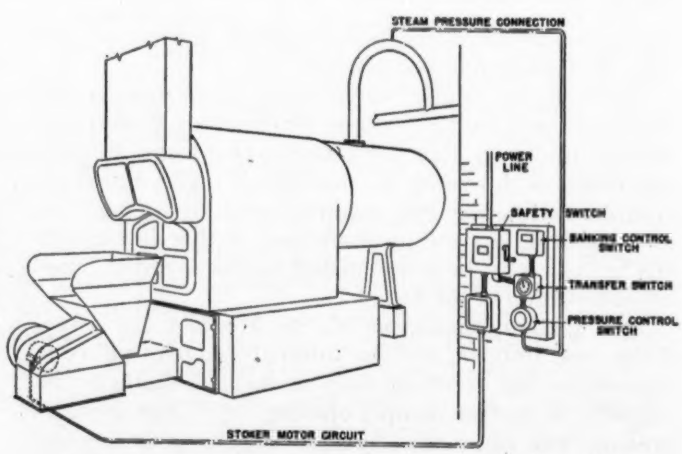


Fig. 2b—"Start-and-stop" control applied to unit stoker

The standard pressure control shown may be changed to or be supplemented by other controls such as aquastat, thermostat, low-water cut-off with or without water feeder and pump control, and "Off-On" time clock.

spring. This controller then increases the forced-draft fan setting. The resulting drop in furnace draft causes the furnace-draft controller to increase the uptake draft, so that equilibrium is re-establsihed at a higher rate of heat input to the boiler.

A variation in the above set-up, as shown in Fig. 5b, is sometimes proposed, but it has been found unsatisfactory for use with single-retort stokers. In this arrangement the "master" energizes the air-flow controller as described in the preceding paragraph. The latter, in turn, re-positions the uptake damper to give a predetermined draft loss and gas flow across the boiler. The resulting change in furnace draft is then corrected by the furnace-draft controller, which adjusts the forced-draft fan damper, thus completing the regulating cycle.

The objection to this system lies in the fact that when the fire doors or ashpit doors are opened, to bar the fuel bed or dump and clean the fire, inrush of the cold air causes the furnace draft to drop somewhat, and because of this drop, the furnace-draft controller reduces the setting of the forced-draft fan, to restore equilibrium. As the steam pressure is quite likely to fall somewhat during the cleaning period, additional coal is fed by the action of the "master," but meanwhile the forced-draft supply has been decreased as the result of the loss of furnace draft; so the steam pressure continues to fall and heavy fires with accompanying clinkers and smoke are likely to result.

Comparing this arrangement with those shown in Fig. 4 it will be noted that in the latter, the uptake draft is directly controlled from furnace draft so that any loss in draft is promptly corrected.

The advantage claimed for this layout (Fig. 5b) is that it prohibits the reduction in furnace draft below that for which the regulator is adjusted and so avoids plus pressure and overheating. This advantage is outweighed by difficulties already described with underfeed stoker operation. However, this type of control has been satisfactorily used with other types of stokers.

In the foregoing it has been necessary to describe the various steps through which the several controllers move, with each change in steam pressure. Actually, the operation of the several controllers is smooth and practically simultaneous, so that equilibrium is restored promptly. The speed and smoothness with which this balance is re-established is the measure of satisfactory performance of a combustion control system, since any undue delay or hunting would be detrimental to overall efficiency.

Instruments Essential

Certain instruments are very desirable, both to facilitate initial adjustment of the controls, and to assist the operators in making the necessary re-adjustments which are generally required, when fuel or load conditions change materially. The furnace-draft gage is generally rated as of first importance, with gages showing uptake draft, and undergrate pressure, in second and third place. These gages are generally of the indicating rather than the recording type as they are primarily for the use of the fireman in making adjustments.

A recording or indicating CO_2 meter, or a steam-flow-air-flow meter, is also essential for good operation. In some cases a smoke indicator or recorder may be useful, although this is not so helpful for underfeed stokers, since smoke is not so quick an indication of the fuel-air ratio as in the case of oil or spreader-stoker firing.

Temperature indicators or recorders showing the exit gas and steam temperatures are also recommended, as they are of value in showing the condition of the heating surfaces, boiler baffles, etc.

The operator who knows the steam output, exit gas temperature, boiler draft loss, and excess air can judge closely the performance of his boiler, and make any necessary adjustments to maintain maximum efficiency. With proper instruction, most firemen soon get to know their instruments and use them to help secure better and easier operation. When properly adjusted the fires quickly respond to load changes and are easy to burn down and clean. In other words, a well-controlled fire is a clean fire, and a clean fire is easy to handle.

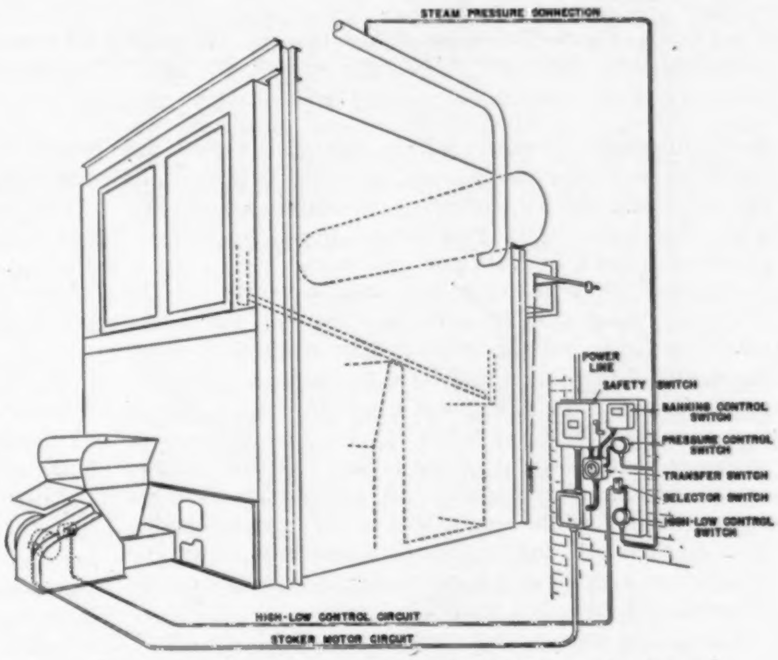


Fig. 3—"High-and-Low" control applied to unit stoker

Note that stoker normally runs at either the high or low feed rate, but if pressure continues to rise at the low rate, the pressure-control switch stops the stoker motor. The alternate controls as listed in Fig. 2b may be used.

Installation and Attention Important

Location of the control elements is important to assure proper operation, as well as to facilitate easy inspection and adjustment. They should never be placed so high that they cannot be conveniently reached from the floor, or in a dark, inaccessible corner, where they will seldom if ever receive attention. Sometimes the installer works on the theory of putting the controllers "out of the way," but it is better to locate them where the fireman cannot evade them, rather than where he will soon forget their existence.

There are too many instances of boiler plants that

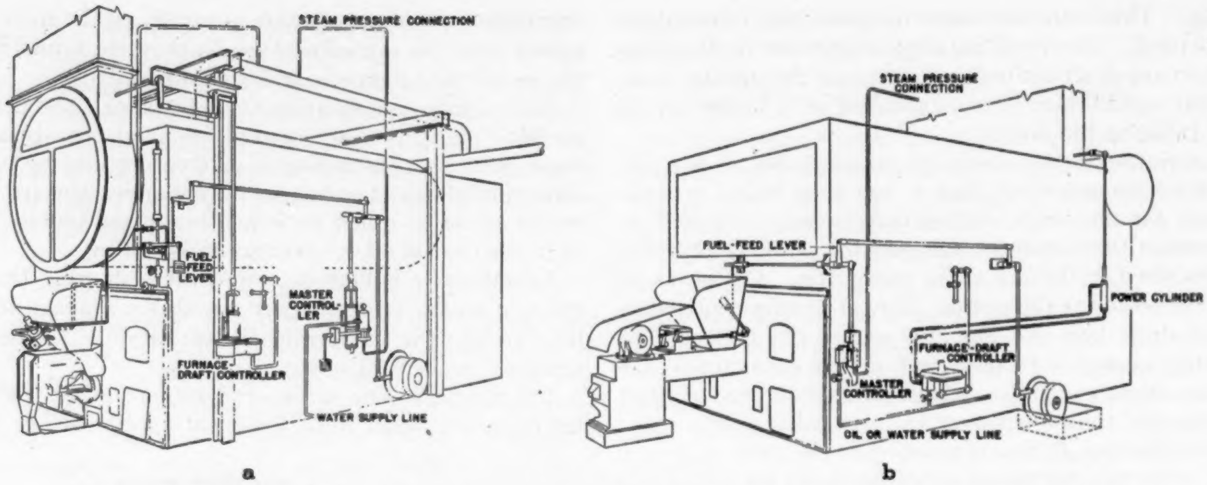


Fig. 4—Two types of "positioning" control applied to stoker—a—steam drive; b—motor drive
The "master" directly controls the fuel feed and forced draft from steam pressure, and the furnace-draft controller regulates the uptake damper.

have automatic controls which upon inspection are found to be totally useless, and in fact a detriment, as the only thing which is working is the coal-feed controller, while the forced-draft and furnace-draft "regulators" remain in a fixed, "never right" position.

Positioning control regulators usually consist of the "master," which combines the pressure diaphragm, and power element, and the furnace-draft controller, which has the diaphragm with its pilot valves separate from the power element. In this case it is usually best to mount the master on the front boiler column, or a convenient buckstay near the front of the boiler, and the furnace-draft controller diaphragm on the real column, where its power element will be convenient to the damper handle. The master and furnace-draft diaphragm should be mounted at a convenient hand height, so that they can be adjusted and inspected readily.

When rigid linkages are used, there should be ample length in the reach-rods, so that side thrust on the power element is minimized. It is seldom desirable to install this type of regulator on a panel-board, or on the wall at a distance from the boiler front, because if the length of reach rods or cables is excessive, it is difficult to avoid sagging and lost motion, both of which are detrimental to good regulation. The use of rigid linkages, rather than exposed cables, is preferable, especially for the horizontal runs, because it is difficult to keep lost motion out of a long horizontal run, unless the cable is supported at frequent intervals. In some types of controllers the cables are carried in pipe, with attached pulleys; in this way sagging is avoided and the cables are protected.

The motor starter is connected to the stoker motor by conduit only, hence its location may be selected for convenience, but at a temperature approximately the same as the motor. The heater element for motor overload protection, which is located in the starter box is designed for a surrounding air temperature of approximately 100 F. If the temperature of the surrounding air is excessive, as sometimes may be the case if the starter is mounted on the boiler, the overload relay may open the motor circuit even though there is no overload on the motor.

When metering controls are used, the controlling and power elements are sometimes separated, with the controllers mounted on a panel, along with the instruments which show the boiler performance, and the power elements mounted adjacent to the levers which they operate. There should then be means for making adjustments in the coal-air ratio, at the panel board, as well as provision for putting the controls on "manual" when desired. These controls frequently have a "station master," which serves to balance the load on the several boilers, as desired.

Many of the power elements provide means for manual adjustment so that if the power source, whether it be electricity, oil or compressed air, should fail, the operators can still adjust the manual control levers and keep the plant running.

Suggestions for Operators

A source of possible trouble with the metering control is the reaction of the controller to a hole in the fire. The free flow of excess air through this hole is likely to cause an increase in the gas flow through the boiler, which will result in an increase in the pressure drop across the air-flow controller diaphragm. As this increased pressure drop was not accompanied by a corresponding movement of the master regulator, the air-flow controller acts to reduce the total air to the fuel bed.

As the forced draft is thus reduced, the steam pressure falls, so that the master increases the rate of coal feed, but without making a proportional increase in the forced-draft supply. If the fuel bed is not attended to by the operator, this condition is likely to continue to become worse, and it is not wise to install a metering control without having a steam flow-air flow recorder, or CO_2 meter for the operator's guidance, so that improper fuel bed conditions can be quickly noted and corrected.

While the response to load swings is quite rapid, there are certain limitations and precautions which must be understood. When the load increases rapidly from a semi-banked, or very low fire, the fuel bed may be so caked over the retort, that an appreciable time will elapse before the heat output will be materially increased. On the other hand, the fuel bed may have burned quite thin, during the light-load period, so that there is little or no fuel on the grates. The feed must therefore be increased sufficiently to build up the fire, and meanwhile, the excess air flow through the grates will be high.

Under these conditions the fireman should reduce the forced draft manually to give the fire a chance to spread out over the stoker; or as an alternate, he may increase the coal feed beyond what the controls are normally set to deliver, to speed up this coverage of the grates. As soon as a full fire has been obtained, the adjustments should be returned to normal position to avoid overloading the fire.

When load increase is in the range where an active fire exists, then the response of the underfeed stoker is very rapid, as the fuel on the grates will generate heat in direct proportion to the increase in air flow. If it is possible to anticipate an increase in load, from a low fire, the operator should manually raise the coal feed and forced draft sufficiently to pep up the fuel bed, but without raising the steam pressure unduly. The stoker will then be ready to respond to the increased demand without undue delay.

The skilled operator can do much to assist the controls by keeping the fuel bed in condition so that it is able to respond when called upon for more or less output.

In the case of a falling load, there are several precautions which the fireman should observe. If the load drops sharply, from a full fire, it may be difficult or impossible to prevent an override in the steam pressure because of the excess heat stored in the fuel bed. Whenever possible the fireman should anticipate this condition and reduce his rate of feed beforehand, so that when the load goes down, the fuel bed will be relatively light, and will lose its excess heat quickly. In this way unnecessary popping of the safety valve will be avoided.

As noted elsewhere, a sudden decrease in the air flow, combined with a heavy incandescent fuel bed, may cause overheating and burning of the grates. The best insurance against this trouble is the maintenance of as thin and porous a fuel bed as is consistent with the load and fuel conditions and a properly adjusted combustion control is most capable of insuring such fuel-bed conditions.

When a light load or banked fire is to be maintained for some time, care must be exercised that the fire does not burn down into the retort. If this occurs, the coal may cake so solidly that the pushers cannot function and also it is quite probable that overheating and burning of the retort sides and fire bars will be experienced.

Certain stokers are provided with bank controls, which avoid this trouble by operating the stoker at regular intervals, so that green coal is forced into the retort, thus forcing the fire out onto the grates.

Stoker maintenance may be greatly influenced by the fuel-bed conditions which, in turn, are under the influence of the combustion-control equipment. For example, if the fuel-air ratio is too high (excess coal) the fuel bed may thicken and cake, so that air flow through the grates and fire is reduced and overheating and burning of the fire bars will occur. Similarly, a sudden reduction in the air flow, because of improper regulator action, will permit the hot fuel bed to radiate heat out to the grate surface, without the counteracting cooling effect of the air, and grate temperatures are likely to rise abnormally.

On the other hand, a low fuel-air ratio (excess air) will result in a thin fuel bed, burned out in spots, so that the grate is partially exposed to the furnace heat and surrounding incandescent fuel bed, with disastrous results.

The properly controlled fuel bed is quite porous and uniform, with the air flow well distributed. This can only be maintained by a well-adjusted control, as the effects of a temporary misadjustment may not be overcome for several hours after the necessary corrections have been made.

Many maintenance problems that have been blamed on improper materials or fuel should properly be charged against poor regulation, and can be corrected most easily by attacking the trouble at its source, which is in the fuel-air ratio and draft control.

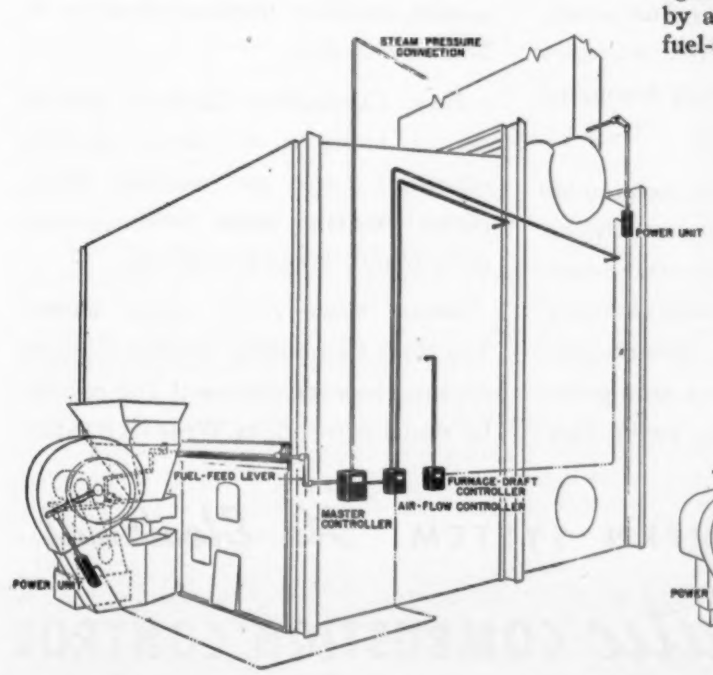


Fig. 5a—"Metering" or "proportioning" control applied to hydraulic-drive stoker

The "master" regulates coal-feed and air-flow controllers, and the furnace-draft controller adjusts the uptake draft.

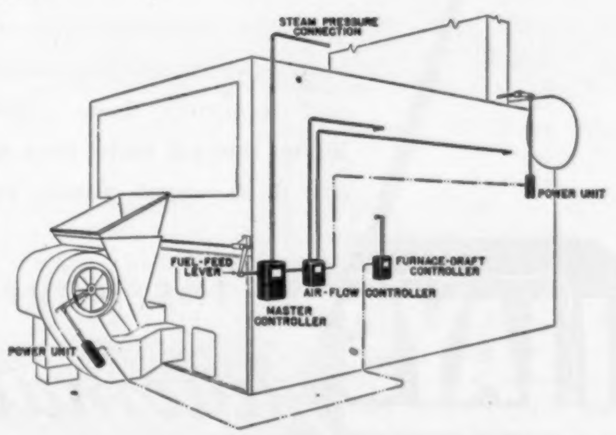
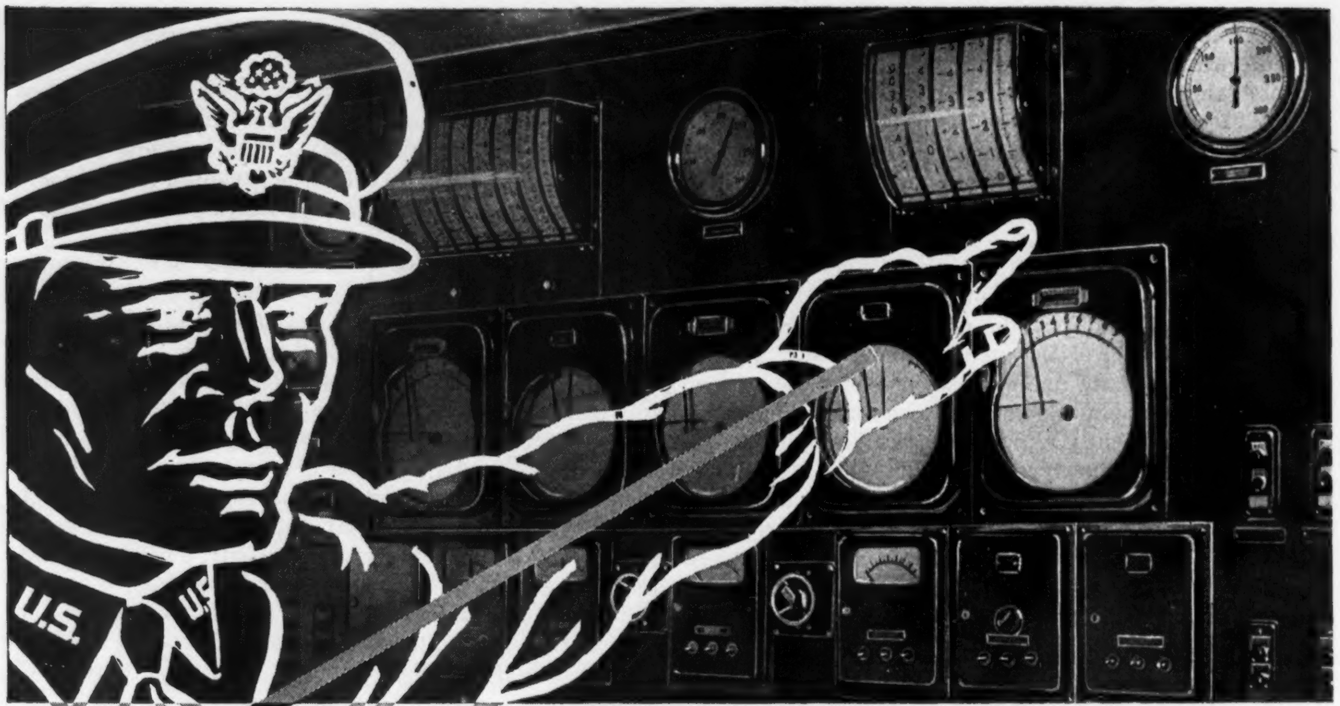


Fig. 5b—Alternate "metering" control applied to integral-fan stoker

The "master" regulates coal-feed and uptake damper levers, and the furnace-draft controller adjusts the forced-draft supply.



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In the earlier installments of this series the general plan for increasing the capacity of the original plant was outlined, and major equipment installed in the first step of the development program was described. This installment, the fourth of the series, deals with the pumps, the main unit condenser and its auxiliaries, and the equipment installed so that steam from the new steam generators can be utilized, when desired, in the old low-pressure section of the plant. The method by which condensate, under these conditions, is returned to the high-pressure section is also described. The final installment, to appear in the May issue, will discuss the subject of piping and give operating results based on twelve months' experience with the enlarged plant.

by H. E. MACOMBER

The Detroit Edison Company

Pumps

THE general arrangement of pumps for handling the condensate and boiler feedwater associated with the new 75,000-kw unit and new steam generators is similar to that used for the most recent units installed at Delray and Conners Creek. Starting at the condenser, the arrangement includes a condenser pump which removes condensate from the hotwell, followed in series by a heater pump which pumps the water through the drains, cooler and the two low-pressure stage heaters to the suction side of the boiler-feed pump. The boiler-feed pump delivers the water through the two high-pressure stage heaters to the steam generators. A fourth pump, installed for heat balance purposes, is used to introduce the heater drains into the condensate circuit on the suction side of the boiler-feed pump. With the exception of the latter pump, all pumps are installed in duplicate and as a group are associated directly with the operation of the new main unit.

Experience has shown that, for quiet operation of a pump removing condensate at saturation temperature from a condenser hotwell, the pump impeller must be located at least four feet below the surface of the water in the hotwell. To meet this requirement with a horizontal-shaft pump it was necessary to provide a pit under the condenser of the new Marysville unit. The depth of the pit is such as to give a submergence of five feet of water on the pump impeller with normal level in the hotwell. Only one such pump was installed in the pit, the general arrangement being shown in Fig. 49. This pump has a capacity of 1570 gpm at 870 rpm and develops 30 psi gage discharge pressure with a suction pressure of 2 to 3 psi absolute. It is driven by an adjustable-speed, 75-hp, 230-volt, d-c motor.

The duplicate-purpose condenser pump, shown at the left of Fig. 49, is a three-stage vertical-shaft pump. It was installed to gain operating experience with this type which has the advantage over the horizontal-shaft type that it can be installed at the condenser-room floor level, thus eliminating the need for a pit. The suction and dis-

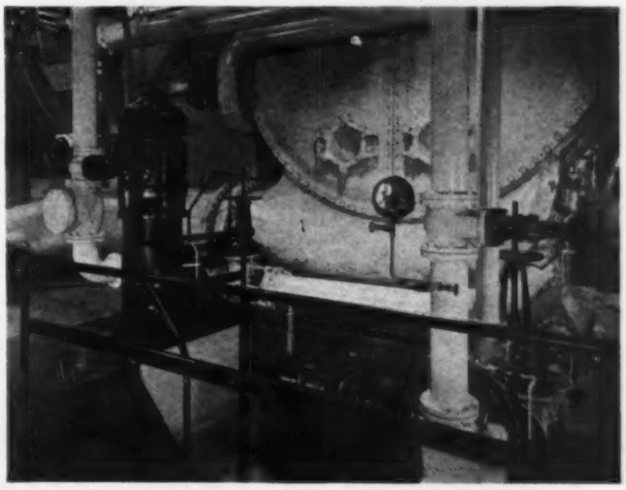

Fig. 49—Condenser pumps for new 75,000-kw turbine-generator

charge connection flanges, 12-in. and 8-in., respectively, are both above the floor level. The pump suction casing, made of 24-in. extra-heavy pipe with a welded closure cap at the bottom, extends 8 ft below the floor level. Entrance to the first-stage impeller suction nozzle is about five inches above the bottom of the casing and the resulting head on the impeller is about ten feet nine inches with normal level in the hotwell. The suction circuit is entirely free from possible air leakage, the only shaft gland being at the top of the pump where it is subjected to discharge pressure. The pump has a capacity of 1560 gpm at 875 rpm and has the same discharge

pressure ratings as the horizontal-shaft pumps. It is driven by a 75-hp, constant-speed, squirrel-cage type, 220-volt, a-c motor.

The heater pump, receiving water from the condenser pump, discharges through the thirteeenth-stage heater-drains cooler and thirteenth- and tenth-stage heaters and delivers the water to the suction of the boiler-feed pump (see Fig. 58). The heater pump and the boiler-feed pump operate in tandem, being driven by the same means and forming what is called the combination heater-boiler feed-pump unit. The heater pump has a rated capacity of 1370 gpm with a discharge pressure of 223 psi and a suction pressure of 28 psi. The boiler-feed pump has a rated capacity of 1455 gpm with a discharge pressure of 1200 psi and a suction pressure of 183 psi when handling water at a temperature of 260 F. The boiler-feed pump discharges through the seventh- and fourth-stage heaters to the boiler-feed header system. The rated discharge pressure of 1200 psi is sufficient to feed water to the steam generators under a pressure condition which would involve blowing of all safety valves.

One combination heater-boiler feed-pump unit, shown in Fig. 50, is driven by a 4800-volt, slip-ring motor rated 1500 hp at 1785 rpm. This unit is used for normal operation. The reserve unit, shown in Fig. 51, is driven by a 1438-hp, 1770-rpm steam turbine designed for 815-psi, 900-F steam at the throttle. The exhaust steam from the turbine is normally delivered to the main condenser but may be exhausted to atmosphere if desired. As is the practice in all the Company plants, the speed of the combination units is adjusted manually for regulation of pressure on the feedwater header.

The heater-drains pump, during normal operation, takes drains from the fourth-, seventh- and tenth-stage heaters and discharges to the suction of the boiler-feed pump where the drains join with the condensate discharged by the heater pump. The drains pump, when handling water at 258 F, has a rated capacity of 287 gpm with a discharge pressure of 210 to 216 psi and a suction pressure of 20 to 26 psi. It is driven by a constant-speed, 220-volt, a-c, squirrel-cage motor rated at 60 hp, 1760 rpm.

In case of outage of the heater-drains pump, drains from the fourth-, seventh- and tenth-stage heaters can be cascaded to the thirteenth-stage heater and passed through the thirteenth-stage drains cooler to the condenser. Although this method of operation increases the heat rate of the main unit by something over 100 Btu per kwhr the anticipated outage of the heater-drains pump is not thought to be sufficient to justify the installation of a reserve unit.

The use of combination heater-boiler-feed pumps operated by the same driving means has been common practice in the Company's power plants for many years. With this type of pump it is convenient, without the the duplication of driving facilities, to use a relatively low-pressure heater pump and low-pressure heaters ahead of the boiler-feed pump and the high-temperature heaters. Experience with this type of equipment has shown it to be highly satisfactory from the standpoint of both operating convenience and economy.

In the earlier installations, up to and including the 400-lb section of the Delray Power Plant, the combination pumps used in normal operation were driven

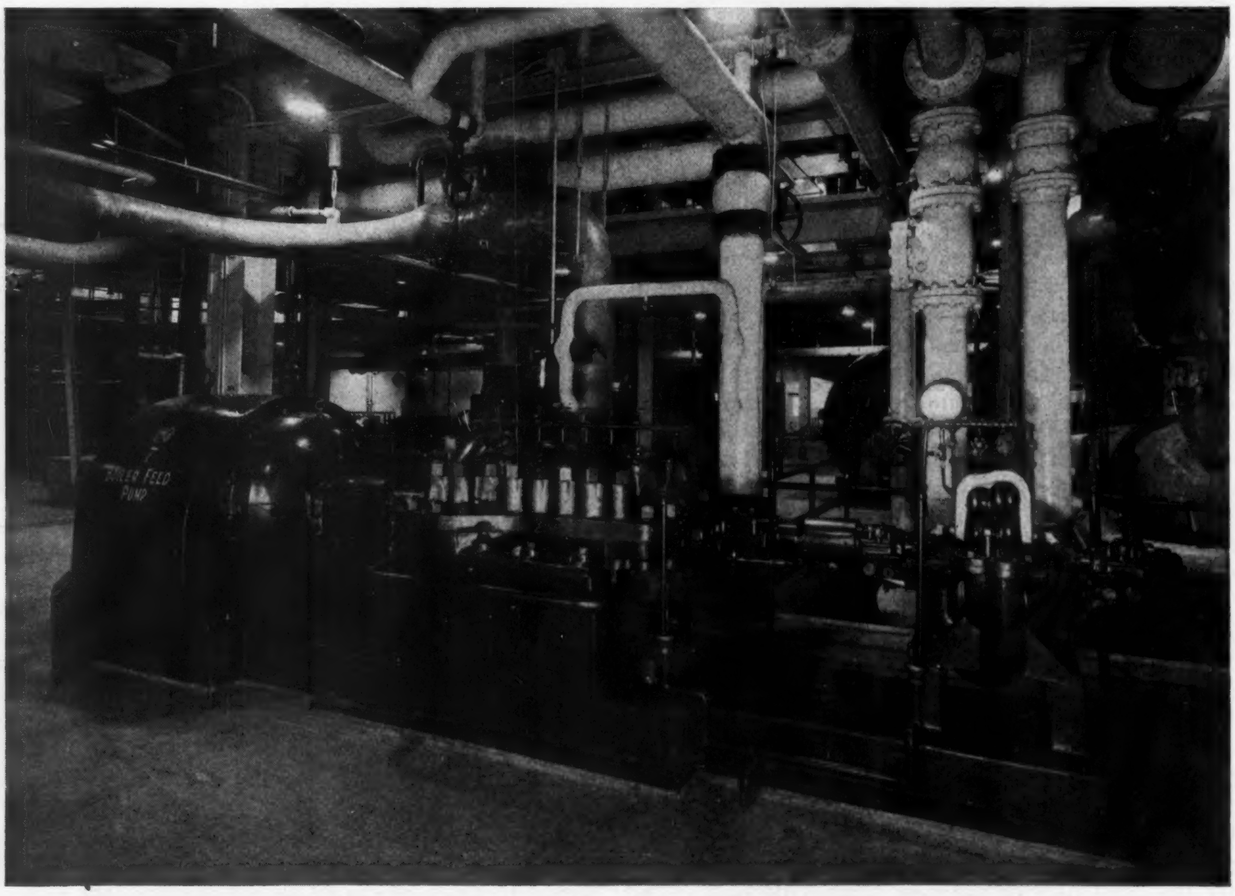

Fig. 50—Electrically driven combination heater-boiler-feed pump unit

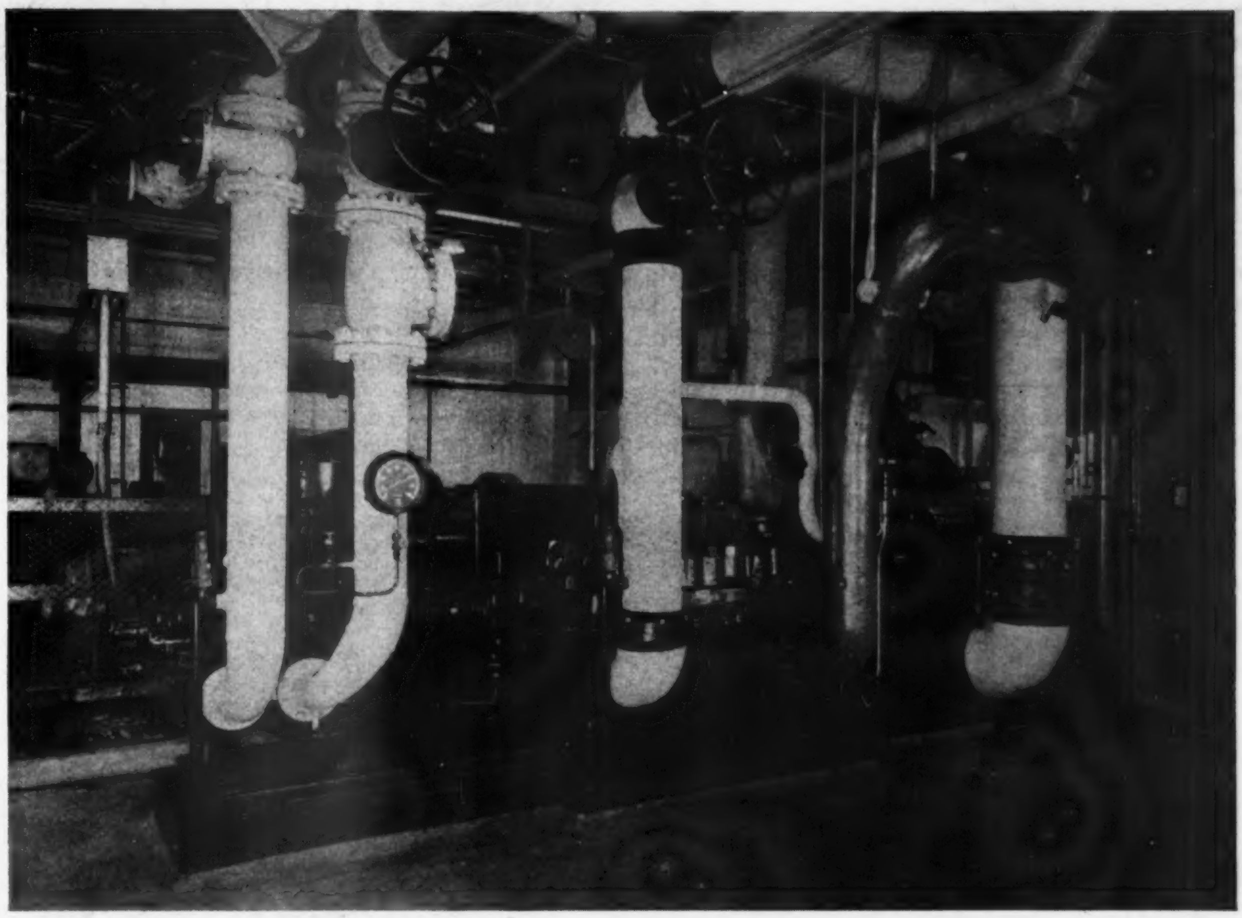

Fig. 51—Turbine-driven combination heater-boiler-feed pump unit

by adjustable-speed d-c motors. For pressures in this range, pump speeds not exceeding 1200 rpm are in order and a direct-current motor of required size can be successfully utilized. Higher pressures, however, require higher pump speeds or the use of exceptionally long and costly multistage pumps. At speeds much over 1200 rpm, d-c motors in the large sizes required have not been considered suitable because of commutation difficulties. Because of this limitation, in the rebuilding of Conners Creek for 600-lb operation, and in the 800-lb section of Delray, a-c slip-ring motors were adopted as the driving means for the electrically operated combination pumps. The reserve units in each case are steam-turbine driven. Thus, the use of the a-c, slip-ring motor for driving one of the new combination pumps at Marysville and a steam turbine for the other follows a Company practice already established. The a-c motors are provided with twenty speed-control points giving a range of speeds from 1450 to 1785 rpm, or about the same number of points as are used in the field controls of the earlier d-c combination-pump motors.

From time to time over a period of years various arrangements for regulating flow and pressure of boiler feedwater have been considered. Among these schemes have been constant-speed drive with throttling control; primary and secondary pumps with speed control on one of the two pumps; all turbine drives; the use of hydraulic and electric couplings; and a step-up gear to allow a pump to be driven by a secondary motor of lower speed. Thus far, none of these methods have been preferred to the arrangements that have been used.

Condenser

The surface condenser serving the new main unit is a single-pass type, designed to accommodate a maximum of 10,796 three quarter-inch OD tubes, 24 ft long. The tube sheets are Muntz metal, 15 ft 3 in. in diameter and $1^1/_2$ in. thick. The tubes are No. 18 gage Admiralty metal rolled into both tube sheets. The 10,252 tubes installed give a total surface of 47,800 sq ft. This number of tubes is 544 less than the number installed in the most recent condenser at Delray serving a turbine of like capacity. The reduction of surface was considered to be justified because condensing-water temperature at Marysville during the summer is six to seven degrees lower than at Delray. The tubes are arranged as a double-folded bank forming two inner air-cooler sections cross-connected so that air and non-condensable gases may be removed from one side of the condenser.

The condenser shell, fabricated by welding $^7/_8$-in. thick copper-bearing steel plate, was manufactured in four longitudinal sections to facilitate transportation. Erection joints are gasketed and bolted. The water boxes are cast iron with flat heads. At the entrance of the circulating-water pipe into the bottom of the inlet water box, there is a series of guide vanes to effect an even distribution of flow into the box. The hotwell is of the bathtub type, bolted to the bottom of the shell and provided with the necessary pump suction openings. A view of the inlet end of the condenser is shown in Fig. 52.

The service weight of the condenser, that is, the weight with water in the tubes and water boxes and 30,000 lb of

condensate in the hotwell, is 611,000 lb. In normal operation about half of this weight is borne by the turbine casing and the other half is supported by four brackets, two on either side of the shell, which rest on springs. One of these spring supports is shown in Fig. 53. It consists of eight double-coil car-type springs resting on a steel block which in turn is supported at each end on removable sole plates. All springs are carefully calibrated and tagged before installation. The sole plates, or shims as they are sometimes called, are machined to an exact thickness that will give the desired divisions of weight between the turbine casing and the spring supports. This division is such that, in the event of loss of circulating water and an emergency exhaust to atmosphere which would cause expansion of the metal parts, a minimum downward load of 20,000 lb on the turbine casing will be assured. When the top half of the turbine casing is removed in order to work on the machine, it is desired that most of the weight normally suspended from the turbine casing be removed. However, to insure against any upward thrust a minimum of 20,000 lb of the condenser weight is allowed to remain. To provide for making the necessary adjustment, each spring support is equipped with a hydraulic jack as shown in Fig. 53. The four jacks are connected to the same hydraulic pump so that they all work together. Operation of the jacks causes further compression of the springs and permits the withdrawal of the removable sole plates and the substitution of others of greater thickness. The thickness of the latter plates is such that, when the jacks are backed off, the weight remaining suspended from the turbine casing will be about 20,000 lb. The hydraulic jacks are also used in the event that it is necessary for any reason to remove the coil springs. In this case the jack screws shown in Fig. 53 are raised to support the condenser after the hydraulic jacks have been raised sufficiently high to relieve the pressure on the sole plates. The plates are then removed so that, when the hydraulic jacks are lowered, the springs will be freed and can be removed from the supports.

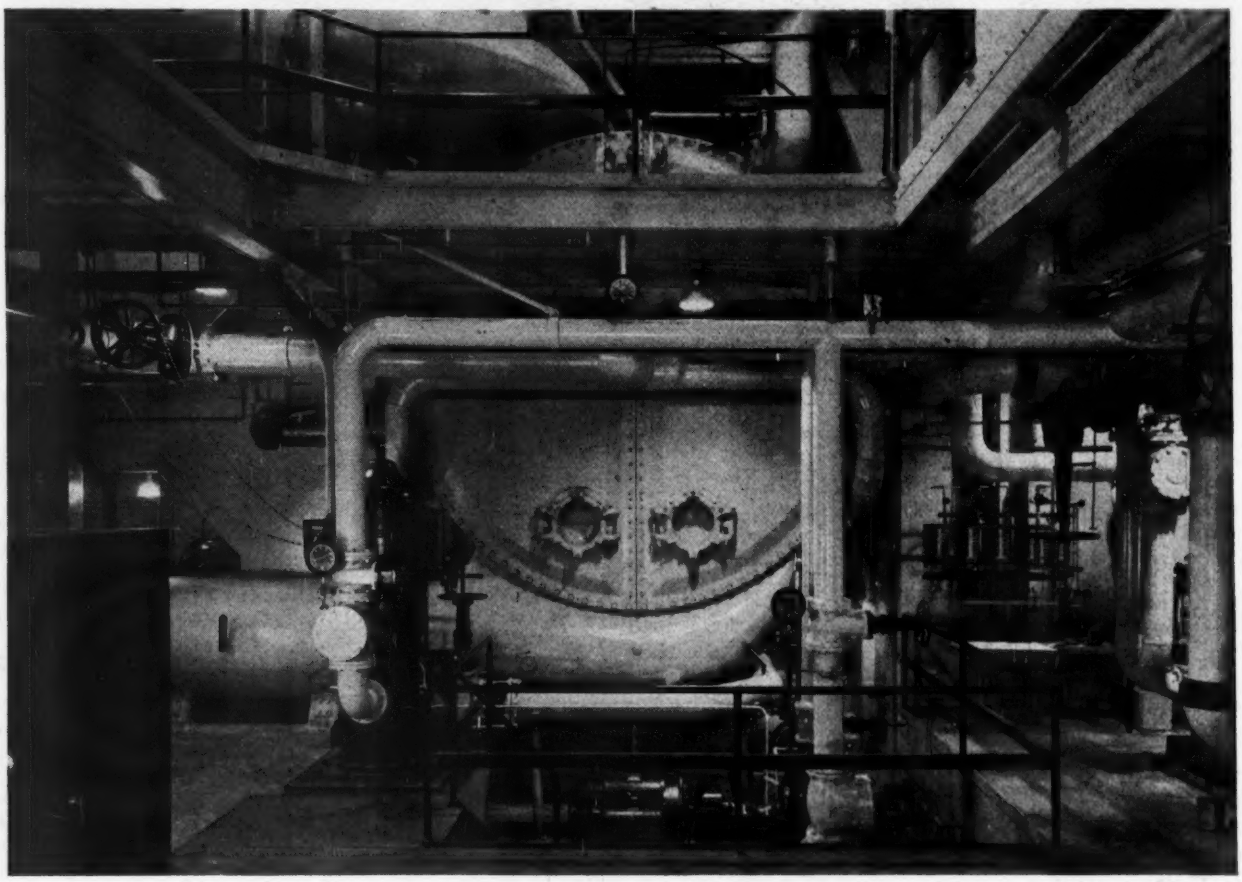

Fig. 52—Inlet end of main unit condenser

Circulating Pump

Condensing water is circulated through the condenser by the 48-in., horizontal, double-suction, split-case volute pump shown in Fig. 54. This pump has a design capacity of 70,000 gpm at a total dynamic operating head of 13 ft of water when running at 240 rpm. It is driven by an adjustable-speed, 240-volt, d-c motor rated 350 hp at 240 rpm, and 116 hp at 166 rpm. Speed adjustment in the range between 240 to 166 rpm is by field regulation; armature control provides for further reduction in speed to 90 rpm.

The Company's present practice of using only one circulator for a large main unit was adopted in 1938 with the installation of the 75,000-kw machines at Delray. Until that time, and starting with the installation of the first 45,000-kw unit at Conners Creek in 1918, all condensers serving turbine-generators of 30,000-kw and larger were equipped with two circulating pumps. There were several factors in the situation at that time which

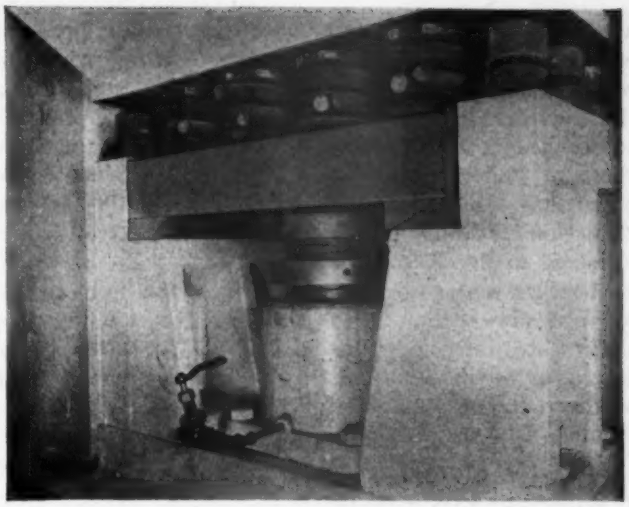

Fig. 53—One of the four condenser supports

were believed to justify this practice. Among them were, the question of reliability of the pumping equipment, the desirability of a broad range in flow adjustment, lower power consumption at partial loads, and smaller dimensions of pumps and connecting piping. Over the years, however, operating records showed that circulating pumps and their drives can be relied upon for long periods of operation without interruption. In addition, modifications in plant cycle, the use of higher steam pressures and temperatures, improvements in turbine design and construction, the more effective distribution of surface within the condenser and more recently the reduction in size of tubes from one inch to three-quarter inch, together with cleaner tubes as a result of chlorination, have very materially reduced the quantity of condensing water required per kilowatt of turbine capacity. In this connection it is interesting to note that the practice with the early machines was to provide pumping capacity of the order of 2.5 gpm per kilowatt of capacity and, as recently as about ten years ago, a value of 2 gpm per kilowatt was used. In contrast, the pumping capacity provided for the latest machines, including the new Marysville unit, is 0.93 gpm per kilowatt of capacity or a decrease of more than 50 per cent. Obviously, the saving in equipment and construction costs by using one pump instead of two is considerable.

To evaluate the effect of the condenser alone in reducing the quantity of circulating water per kilowatt of capacity, it is proper to compare the pounds of cooling water to the pounds of steam entering the condenser. For the early machines this ratio was of the order of 122, that is, sufficient pumping capacity was installed to supply 122 lb of circulating water for each pound of steam to be condensed in the condenser. In the latest machines, including the new Marysville unit, this ratio has dropped to 75, showing about a 40 per cent decrease. The influencing factors here are the more effective distribution of surface in the condenser, the use of $^3/_4$-in. in place of 1-in. tubes and the chlorination of the circulating water.

Air Pump

Removal of air and non-condensable gases from the condenser is accomplished by means of a horizontal rotating-type reciprocating vacuum pump shown in Fig. 55. This particular pump was initially installed in 1921 at the Delray Plant to serve one of the earlier 30,000-kw turbine-generators. It is a single-cylinder, two-stage pump and has a displacement of 2050 cfm when running at 100 rpm. The pump is driven by a shunt-wound, 230-volt, d-c motor having a speed range of 100 to 30 rpm.

It had been the practice of the Company to use reciprocating vacuum pumps on all main condensers until the last unit of those presently installed at Conners Creek was put in. In this case and also in the later installation of the three 75,000-kw units in the high-pressure section of Delray, steam-jet air ejectors were used. The decision to use ejectors in place of reciprocating pumps was partly a matter of lack of space to install anything but a vertical type reciprocating pump and partly due to the fact that reciprocating pumps of the type desired were not readily available.

It is believed that both the reciprocating air pump and the steam-jet air ejector have desirable features. The former has the advantage of greater convenience when starting the turbine and will permit a quicker start than the ejector unless the ejector is equipped with very large nozzles for starting purposes. Also, the reciprocating pump has the advantage that all air and gases removed from the condenser are discharged to atmosphere; whereas, with the ejector there is the possibility that gases, such as ammonia and carbon dioxide, may be carried back to the condenser and into the condensate circuit with the condensed ejector steam. The ejector, on the other hand, requires less space and it is not necessary that it be located in a conspicuous place for ready observation by the operating personnel. Moreover, the ejector has a much lower first cost. Against that, however, is the fact that the overall operating cost of the ejector is higher than that of the reciprocating pump so that, considering fixed charges and operating costs together, the two types of equipment are about on a par. On the basis of operating experience within the Company, it may be said, that, all things being equal, the preference is for the reciprocating type pump and the probability is that, when additional main units are installed, they will be equipped with such pumps.

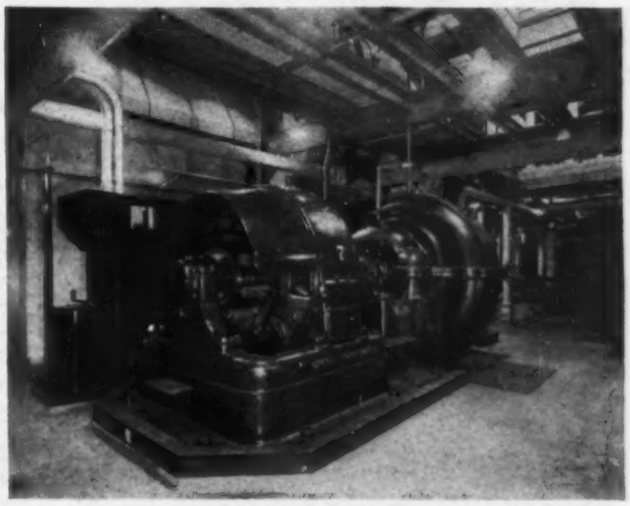

Fig. 54—Circulating water pump

Combined Operation of High- and Low-Pressure Sections

Removal in the early stages of the project of an old 10,000-kw turbine-generator, as explained in a previous article, reduced the low-pressure generating capacity of the plant by about six per cent. However, removal of two of the low-pressure boilers to make way for new steam generators reduced the low-pressure boiler capacity by about twenty-five per cent, leaving the plant with insufficient low-pressure steam for simultaneous operation of the remaining low-pressure turbines.

Transfer of Steam

To compensate for this deficiency, a pressure-temperature reducing station, consisting of two elements in parallel, was installed whereby steam from the new high-pressure steam generators can be fed into the low-pressure main steam header, thus making it available, along with steam from the remaining low-pressure boilers, to all of the low-pressure turbines. With the reducing station in operation it is possible to load the plant to the full capacity of its high- and low-pressure turbine-generators. In addition, the reducers provide for considerable flexibility in the combined operation of the high- and low-pressure sections.

Each reducing element has a design capacity of 279,000 lb of steam per hour at a final condition of 275 to 300 psi and 700 to 725 F. For this capacity the required input to the element is, (*a*) 246,500 lb of steam per hour, having a pressure of 825 to 875 psi and a temperature of 900 to 925 F, and (*b*) 32,500 lb of water per hour at a temperature of 375 F. Of the above amount of steam, 6500 lb per hour is used to atomize the water. Water supplied to the reducing elements is taken from the high-pressure boiler-feed header. Either element can be operated alone or the two can be operated together in which case the combined capacity is 558,000 lb of steam per hour. Both elements are equipped with automatic de-superheating control and manual pressure control and, in addition, one is equipped with automatic pressure control. In normal operation, when steam from the reducing station is being used to supplement steam from the low-pressure boilers, pressure in the low-pressure steam header is controlled by regulation of the low-pressure boilers in the usual manner. Under these conditions, the automatic pressure regulator at the reduction station is cut out and the reducing valves are adjusted manually to control flow only. In the event that steam were to be transferred from the high- to the low-pressure section with none of the low-pressure boilers steaming, pressure in the low-pressure header would be regulated by the automatic pressure control at the reducing station. A diagram of one of the elements is shown in Fig. 56 and a photograph in Fig. 57.

Transfer of Condensate

Obviously, the delivery of steam from the high- to the low-pressure section required that some means be provided for the return of an approximately equal quantity of condensate to the high-pressure section. This might have been reasonably simple had it not been for the fact that the condensate in the low-pressure section had too high an oxygen content to be safe for use at the pressures and temperatures existing in the new steam generators. Another problem was that condensate in the low-pressure section is heated to only 215 F before entering the low-pressure boilers; whereas, the boiler feedwater temperature in the high-pressure section is 375 F.

After considering various means for accomplishing the desired result, it was finally decided to return condensate from the condenser of the 50,000-kw low-pressure turbine-generator to the high-pressure section in amounts corresponding to the amount of steam being delivered through the reducing station. This is the largest and newest of the low-pressure units. There were several reasons for this decision. For one thing, this machine, under high load conditions, could supply a sufficient

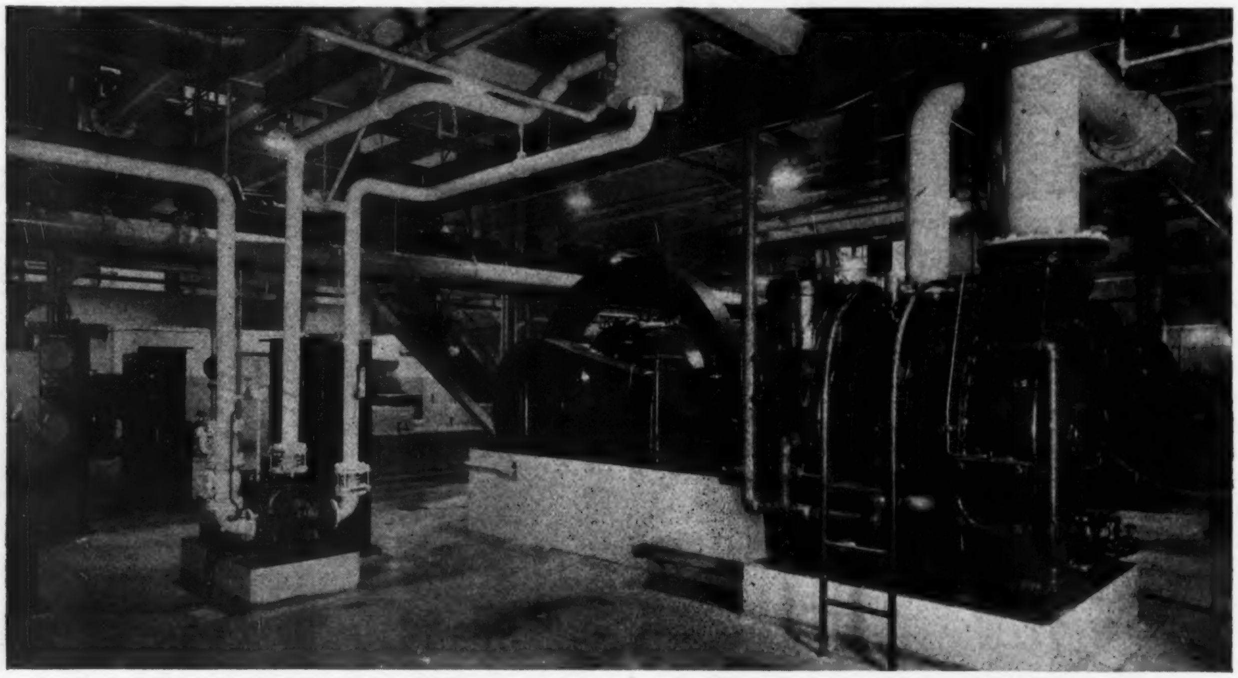

Fig. 55—Reciprocating vacuum pump for removal of noncondensable gases from main condenser

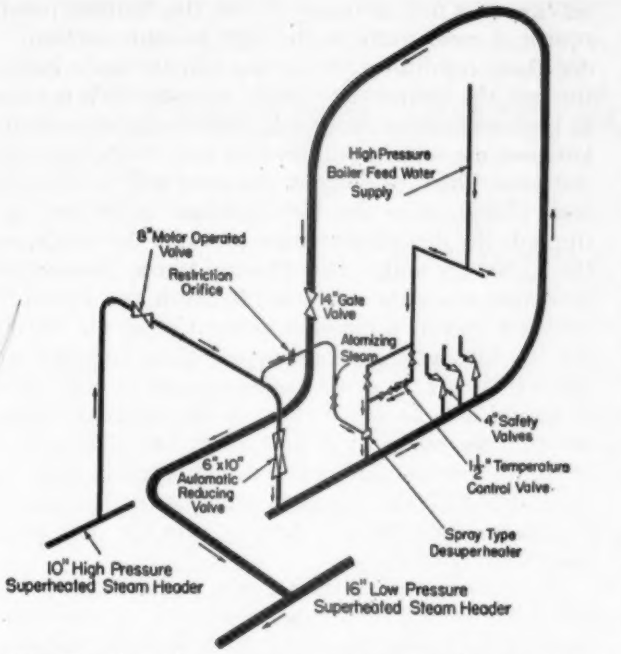

Fig. 56—Diagram of pressure-temperature reducing station

quantity of condensate to balance the maximum capacity of the reducing station. Also, the condenser serving this unit, with slight modifications, could be made a suitable means for deaerating the condensate. In addition, there was available at the twelfth-stage of the turbine an unused extraction opening which could be utilized for a new heater to increase the temperature of the condensate. The limitation that the transfer of steam from the high-pressure section would be contingent upon the operation of the 50,000-kw unit was not unreasonable because normally, with this unit out of service, the quantity of steam available from the low-pressure boilers will be sufficient for the operation of the remaining low-pressure turbines.

It should be remembered that steam from the reducing station is fed into the low-pressure steam header from which it may find its way, not only to the 50,000-kw machine, but to any of the low-pressure machines that may be operating. However, it is only condensate from the 50,000-kw machine that can be pumped directly to the high-pressure boiler-feed system. The pump provided for this purpose is an electrically driven combination heater-boiler-feed pump, duplicate of the electrically driven combination unit installed in connection with the new 75,000-kw unit. It is referred to as the transfer pump.

Flow Diagram

A flow diagram of the high- and low-pressure sections of the plant combined, and including the new twelfth-stage heater and the transfer pump, is shown in Fig. 58. From this it is seen that drains from the twelfth-stage heater, when in use, cascade to the sixteenth-stage heater. The combined drains from the nineteenth-stage heater are returned to the condensate circuit at a point between the nineteenth- and sixteenth-stage heaters. To handle the increased drains resulting from the addition of the twelfth-stage heater, the motor of the drains pump was replaced with one of higher speed and larger capacity.

Originally, there was no automatic level control on the hotwell of the 50,000-kw unit. Condensate was pumped through the heaters, where it attained a temperature of around 200 F, and discharged into the low-pressure boiler-feed suction header on which the boiler-feed storage tanks "float." Occasional flashing at the condenser pump and surging in the heater circuit were not troublesome because pressure on the boiler-feed suction header was maintained by water in the boiler-feed storage tanks. However, the installation of the transfer pump in series with the condenser pump, as shown in Fig. 58, required that the flow of condensate from the condenser pump be uniform and also that pressure at the suction of the transfer pump be sufficient under all conditions of operation to prevent flashing which might vapor-bind both the heater and the boiler-feed pump units. To assure uniform flow, the hotwell was equipped with a low-level float-operated valve. The function of this valve is to admit condensate from the condensate storage system into the condenser when there is a tendency for the level in the hotwell to fall below a predetermined value. To provide for proper deaeration of this condensate, the condenser was equipped with spray pipes similar to those in the condenser of the 75,000-kw unit which were previously described. Maintenance of the desired pressure at the suction of the transfer pump was accomplished by the installation of constant-pressure valves in the circuit as shown in Fig. 58.

The rate of return of condensate to the high-pressure section at any time is dictated by the flow of steam at that time from the reducing station. For operating purposes, a steam-flow and a water-flow meter are provided to show the rate of exchange of steam and condensate between the two sections. The flow of condensate is made to match the flow of steam by manual regulation of the speed of the transfer pump. Thus, the rate of return of condensate to the high-pressure section is not a function of the throttle flow to the 50,000-kw unit. At times, therefore, the flow of steam to that machine may exceed the flow of condensate back to the high-pressure section and again it may be less than the demands of the transfer pump.

In the former case condensate, in excess of that required by the transfer pump, passes on to the low-pressure boiler-feed storage tanks or to the condensate storage system. To effect the desired disposition it was necessary to install a high-level float valve in the boiler-feed storage tank system and a high-level float in the

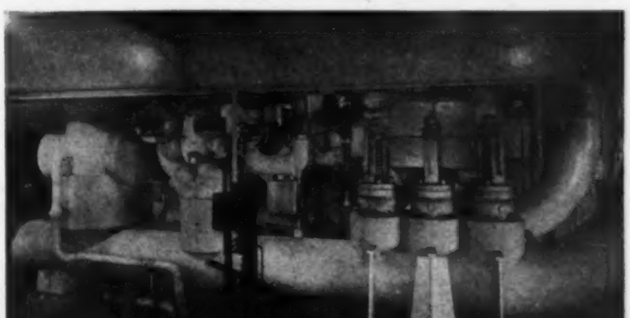

Fig. 57—View of pressure-temperature reducing station

hotwell of the 50,000-kw unit. The sequence of operation is as follows: Condensate not taken by the transfer pump will flow into the boiler-feed storage tanks until they become full and the flow is stopped by the closing of the float-operated valve. This causes the level in the hotwell to rise until the high-level float opens a valve in the connection leading from the condenser pump discharge to the condensate-storage system. It will be noted from the flow diagram that under this latter condition the excess condensate does not pass through the heaters.

When the rate of return of condensate to the high-pressure section of the plant is greater than the flow of steam to the throttle of the 50,000-kw unit, the level in the hotwell will, of course, tend to fall. This causes the low-level float valve to open and admit water from condensate storage to the condenser at a rate sufficient to satisfy the demands of the transfer pump.

Feedwater Heating

Additional heat supplied by the twelfth-stage heater to condensate being returned to the high-pressure section is sufficient, at full load on the machine, to raise its temperature to about 305 F. While this is 70 deg F less than the normal temperature of the high-pressure boiler feed, its effect upon the capacity of the high-pressure steam generators under most operating conditions is not of great consequence. If the temperature of all water supplied to one of the high-pressure steam generators were dropped from 375 to 305 F, it is estimated that its continuous full-load rating without exceeding the design steam temperature would drop from 440,000 to 400,000 lb per hour. An incidental advantage of the twelfth-stage heater is that, under favorable conditions, the thermal efficiency of the 50,000-kw machine may be increased by as much as 250 Btu per kwhr when the heater is in use.

Other Operating Provisions

It is interesting to note that under certain conditions of combined operation, with the 75,000-kw unit in service, it is not necessary to use the transfer pump for return of condensate to the high-pressure section. Under these conditions the return can be made indirectly through the condensate-storage system which is common to both sections of the plant. Excess condensate in the low-pressure section will find its way to storage and, at the same time, equivalent amounts will be introduced from storage into the high-pressure boiler-feed circuit through the deaerating spray pipes in the condenser of the 75,000-kw unit. It is planned to use this method of operation when the exchange between the two sections does not exceed a nominal amount of about 100,000 lb per hr, although it is estimated that amounts up to 200,000 lb per hr could be exchanged in this manner. It should also be noted that, in the event of failure of the transfer pump at a time when the 75,000-kw unit is not in service, the steam-driven combination pump associated with the new unit could be used to introduce condensate from storage directly into the high-pressure boiler-feed header.

The performance of the transfer pump, when handling less than 200,000 lb of water per hour, is unstable because of the characteristic of the boiler-feed pump unit. To obtain stable operation at the lower delivery rates, a $1^1/_2$-in. line provided with a valve was installed from the discharge of the boiler-feed pump back to its suction. By allowing water to recirculate through this line the quantity of water handled by the pump can be kept above 200,000 lb per hr while the actual delivery from the unit is less than that amount.

A two-inch line leading from the discharge of the unit to the condensate-storage system was also installed. The purpose of this line is to avoid the necessity for intermittent operation of the pump when it is desired to use this unit as the source of feedwater in starting up one of the new steam generators. During the approximate six-hour period required to place one of these units in service from a cold start there are frequent but not continuous demands for water. By opening the valve in the discharge to storage it is possible to keep the pump running continuously during the starting-up period without the danger of overheating it.

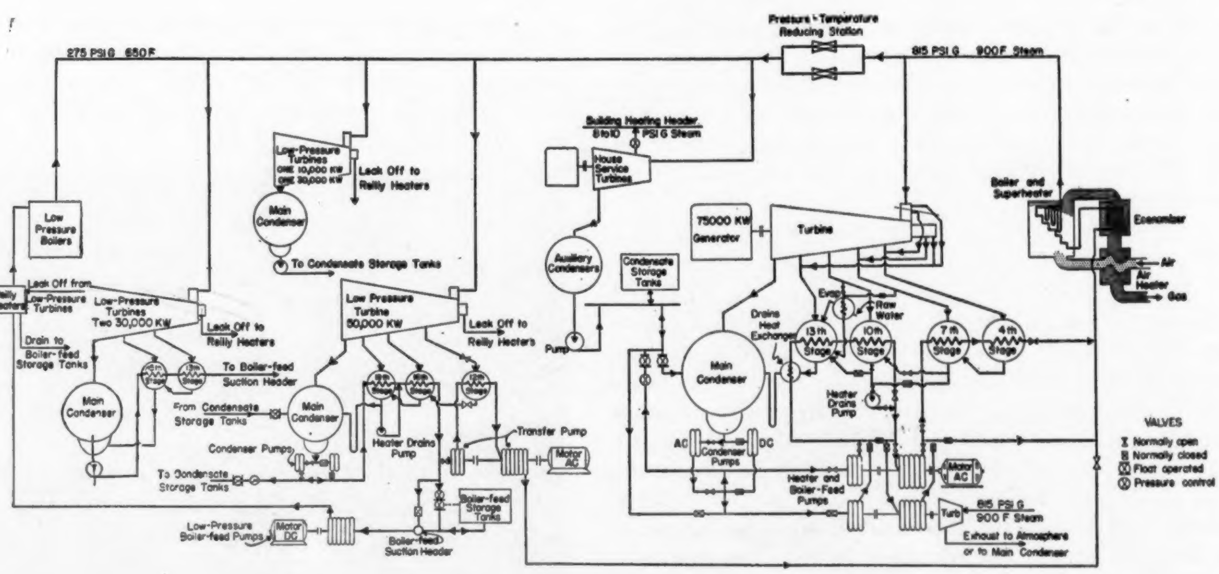

Fig. 58—Flow diagram for high-and low-pressure sections of the plant

Design Data for Overfire Jets

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Results of this investigation were first given in a paper by the author before the Smoke Prevention Association at Pittsburgh last June which was reported briefly at the time by COMBUSTION in its account of the meeting. Subsequently, it formed the subject of a paper, giving full details of the work, at the December 1943 Annual Meeting of the A.S.M.E. These created wide interest among engineers which led the author to prepare the following abridged report of the results, omitting descriptions of the tests and laboratory set-up, and incorporating some heretofore unpublished matter.

IN A recent investigation[1] conducted at Battelle for Bituminous Coal Research, Inc., the performance of steam and fan-driven jets suitable for the introduction of air and the promotion of turbulence over a fuel bed was measured. The optimum values of some hitherto uncertain design features of steam jets were determined. It was concluded that for maximum air delivery the most important feature of design was to make the air tube at least five but not more than ten diameters long. Next in importance was to center the steam nozzle accurately about one air-tube diameter back of the throat. A rounded or funnel-shaped entry to the air tube gave about ten per cent better performance than a square-edged entry. The effects of steam pressure, steam nozzle and air-tube diameter were explored over a range of practical values, but it was concluded that there was no optimum value for these. Furthermore, these factors were found analytically and experimentally to be intimately related to the penetration of the jet, which in the practical case must be related in some undetermined manner to the conditions in each furnace.

What follows is an attempt to establish those relationships for use in jet design. The fair agreement which will be shown between this method and others developed through experience indicates that the analysis has some merit. Certainly there will be exceptions to which it will not apply because of non-uniformity of fuel bed. Perhaps by intelligent modification, some of the curves developed here can be applied to these exceptions. Some others will defy all such rationalization. Mastery of these will depend, as always, on the skill of the individual installers and operators.

Penetration

The velocity patterns of a number of fan-driven and steam-driven jets have been measured when discharging into an open laboratory. Two conditions there differed from those in a furnace: First, the temperature of the surrounding gases was low, and second, their velocity was negligible. However, these unreal conditions were tolerated because of the great difficulty of making such measurements accurately in a furnace.

Numerous facts were determined which seem verified by actual installations. The spread of a jet was found to be about one-tenth of its penetration. Thus, for an effective penetration of 10 ft, the spread is about one foot, and the centerlines of the jets should be approximately one foot apart over that area where overfire air or turbulence is needed. Similarly, if the distance to be penetrated is 5 or 15 ft, the jet spacing should be 6 or 18 in., respectively. Carroll[2] has described some installations on chain-grate stokers in which jets are operating satisfactorily with spacing of this order.

Another conclusion was that for a fixed outlet diameter, the penetration is directly proportional to the weight of air flowing. This agrees with practical experience.

The influence of some other factors, however, remained obscure. For example, although the spread was about one-tenth of the penetration, the precise value of either of these could not be determined because the minimum effective velocity was not known. For the sake of convenience and because it seemed approximately the desired value, the useful limit of the jets was assumed to be the distance at which their velocity had dropped to 1000 fpm. In practice, the burning rate will have an important effect in determining the minimum effective velocity. Only actual experience can show if 1000 fpm is a practical approximation.

Factors Affecting Choice of Jets

In the Appendix the equations relating to the many variables which affect overfire jets are developed. It is

[1] "Overfire Air Jets," R. B. Engdahl and W. C. Holton, *Trans.* A.S.M.E., vol. 65, no. 7, pp. 741–754, 1943.

[2] "Modern Applications of Overfire Air," H. C. Carroll, *Trans.* A.S.M.E., vol. 65, pp. 73–86, 1943.

shown that when certain simplifying assumptions are made the fundamental factors become the burning rate, ratio of overfire air to theoretical air requirement and depth of the region to be penetrated. To be precise, the effects of furnace gas velocity and temperature should be included, but no data are available on these. Both are related primarily to the burning rate but also somewhat to individual furnace construction. Neither influence can be expressed simply.

The design can usually be based on the maximum normal burning rate. The depth of penetration can be equal to the grate length for uniform fuel beds or to the length of the active devolatilization region for a non-uniform bed. The overfire air ratio must be assumed.

Amount of Overfire Air

The percentage of overfire air should obviously be no greater than necessary in order that excess air may be held as low as possible. Often it is observed that in many furnaces no additional air is needed; plenty of excess air is present, and it needs only to be mixed with the combustible. Steam jets alone with no air at all can be used in such cases. Very small amounts of air supplied at equally high pressure can be used to the same effect without imposing the loss due to the latent heat of the steam, but steam is usually cheaper than air at such pressures.

The recommendations of percentage overfire air from various operators range from 5 to 40 per cent of the total air supply (10 to 80 per cent of theoretical if the CO_2 is 9 per cent, corresponding to 100 per cent excess air). More than 30 per cent of the theoretical should not be used unless gas analyses or severe smoking call for more. Later it will be seen from the charts that, for overfire quantities of less than 10 per cent of theoretical, the air pressures required of fan-driven jets are excessive. There is no such lower limit with steam-driven jets.

Jet Selection Curves

Figs. 1 and 2 show curves based on the equations developed in the Appendix. The curves are for a coal having a theoretical air requirement of 10 lb per lb of coal and for overfire air at 70 F and 29.92 in. Hg. Thus, if the grate length (or length to be penetrated) and the burning rate are known and a choice is made of percentage overfire air to be delivered, the diameter of jet is automatically determined from Fig. 1. The weight of air per jet and the jet spacing are then found from Fig. 2 and Table 1, respectively. For fan jets the jet pressure is also found from Fig. 2.

STEAM-AIR JETS—If the air is to be delivered by steam jets, the size of the steam nozzle can be obtained from Fig. 3 which is based on the laboratory data. Fig. 4 shows the steam requirements for each nozzle based on Grashof's equation.

STEAM JETS—If steam jets alone without air are to be used, the nozzle size may be determined approximately by using the same nozzle as specified for 10 per cent overfire air. This is permissible because it was found that the penetration of a bare steam jet is only 10 per cent less than that of the same nozzle in a steam-air jet. If this approximation is not sufficiently accurate, it is a

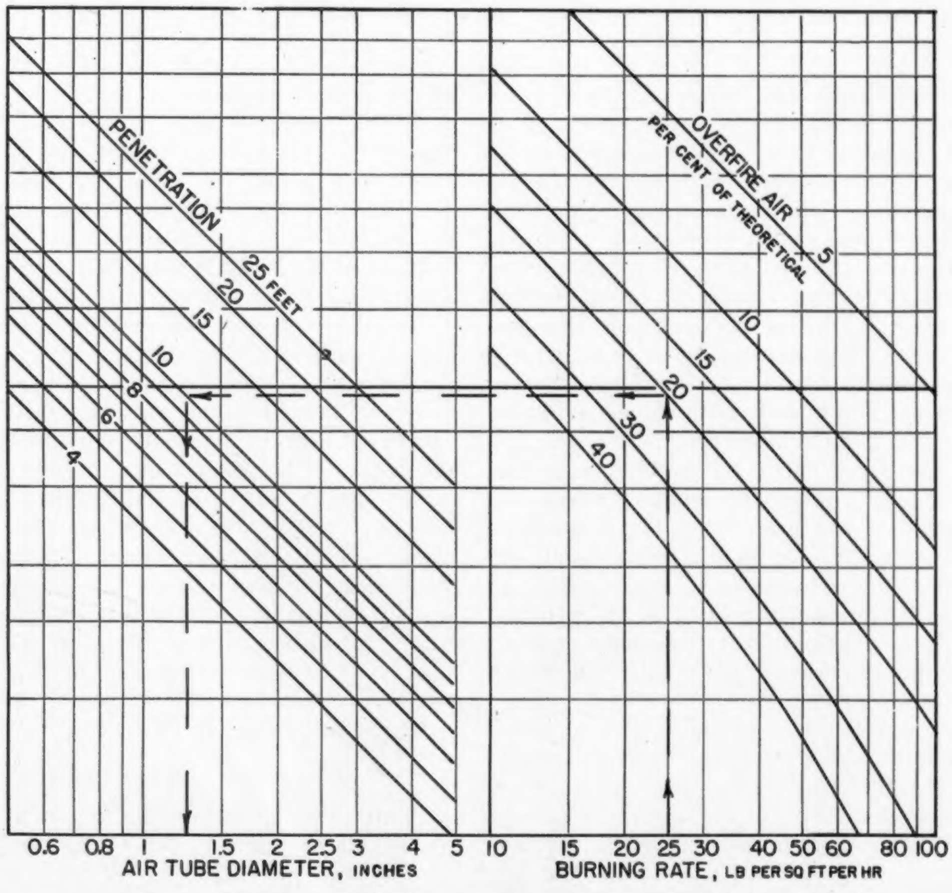

Fig. 1—Chart for calculating air-tube diameter

simple matter to increase or decrease the steam nozzle diameter or steam jet pressure as needed.

STEAM-FLUE-GAS JETS—Steam jets can be used to inject hot flue gas over the fire when there is sufficient excess air. The waste heat goes to supply the latent heat of vaporization of the droplets in the steam jet, and the thermal advantage of hot gas over cold air is obvious. The performance of steam jets handling hot gas is not known. However, penetration, not gas weight, is the primary consideration with these. It is probable that the outlet velocity of the jet is not increased much by temperature, and since penetration depends fundamentally on momentum, temperature should not affect greatly the penetration of a steam-gas jet.

TABLE 1—RECOMMENDED CENTERLINE SPACING OF JETS
(To nearest inch)

Penetration Length, Ft	Jet Diameter, In. 0.5	1	1.5	2	3	4
4	5	5	6	7	8	9
6	7	7	8	9	10	11
8	10	10	11	12	13	14
10	12	12	13	14	15	16
12	15	15	16	17	18	19
14	17	17	18	19	20	21
16	19	19	20	21	22	23
18	22	22	23	24	25	26
20	24	24	25	26	27	28
25	30	30	31	32	33	34

Practical Comparisons

Wassall[3] has proposed an empirically developed method for selection of fan jets and has presented a curve of recommended fan-jet pressure versus length of fuel bed. Although this curve does not include the effect of diameter, spacing, or burning rate, the recommendations which accompany it automatically limit each of these factors to a practical range. Reasonably good agreement is obtained between Wassall's method and the present one. For example:

Assume a uniform fuel bed 15 ft long and 15 ft wide burning at the rate of 50 lb per sq ft per hr with 50 per cent excess air. If the theoretical air required is 10 lb per lb of coal, the total air supplied must be 168,750 lb per hr. Wassall recommends that, in general, the overfire air should comprise from 5 to 15 per cent of the total and that the jets should be 8 to 12 in. apart. His curve of jet pressure specifies 7 in. water for this 15-ft grate. An overfire quantity of 10 per cent (15 per cent of theoretical) and a spacing of 12 in. will be used. This requires 13 jets supplying a total of 16,900 lb of air per hr, or 288 cfm per jet for air at 70 F and 29.92 in. Hg. The 7-in. jet pressure will produce a velocity of 10,610 fpm; hence, each jet must have an area of 0.0271 sq ft. This is supplied by a $2^1/_4$-in. jet diameter.

If the jet spacing had been assumed as 18 in., the number of jets would have been reduced to 9 and the jet diameter increased to $2^{11}/_{16}$ in.

For the same furnace, Figs. 1 and 2 yield the following compared to Wassall's method:

	Figs. 1 and 2	Wassall
Jet diameter, in.	2.5	$2^1/_4$ to $2^{11}/_{16}$
Jet spacing, in.	20.5	12 to 18
Jet pressure, in. water	7.5	7

Some combinations of conditions have been found to result in considerable disagreement between the two methods, partly because the Wassall recommendations do not permit much latitude with regard to spacing, and perhaps partly because the assumptions made in the present method are too restrictive.

For the conditions of the example, Weber[4] recommends a pressure of 15 in. of water. The pressure to be used, of course, depends upon the percent of overfire air selected. The proposed method will agree with Weber if the percentage is reduced in the example from 10 to 6.6 per cent.

Orientation of Jets

SPACING—If allowed to scrub or impinge on the hot furnace walls, the jets may cause serious erosion. Those jets nearest adjacent walls should be spaced from those walls by $^1/_2$ to 1 jet space as given by Table 1. For large chain-grate stokers, Wassall recommends 18 in. as a minimum distance from the walls and suggests turning these side jets away from the walls 6 to 8 deg.

[3] "The Use of Overfire Air in Chain Grate Stoker Operation," Warren W. Wassall, *Solid Fuel Engineer*, vol. 3, no. 1, pp. 5–8, October 1943.

[4] "Boiler Furnace Design for Smokeless Operation," Herman W. Weber *Solid Fuel Engineer*, vol. 2, no. 12, pp. 5–7, September 1943.

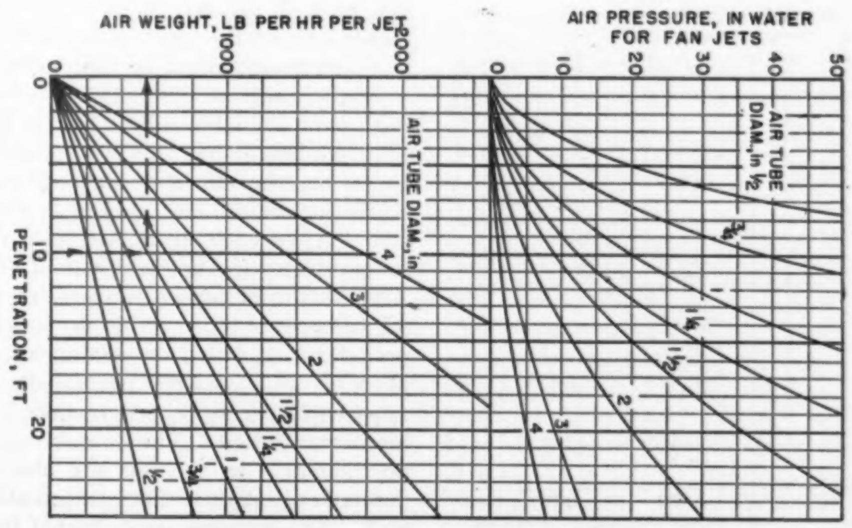

Fig. 2—Chart for air weight and fan pressure

LOCATION—No specific rule for location of the jets can be given. Obviously, they should be directed toward the region which appears to need air or mixing. Preliminary exploration of the furnace by means of a jet from a simple nozzle on a flexible steam hose will often reveal the best jet location. If such a region can be reached by jets from opposite walls this should be better than jets in just one wall. Where such opposed jets are used it seems better to stagger them than to arrange them to meet head-on. Where possible, jets in the bridgewall directed forward give maximum mixing because they run counter to the natural flow. However, maintenance and inspection of jets in the bridgewall may be expensive and difficult.

ANGLE—Jets directed downward probably give the best mixing, but they should not be directed at the fuel bed, as clinkering may result. The downward angle probably results in deflecting some fly ash into the bed which otherwise might pass on through the boiler.

Non-uniform Fuel Beds

Frequently, with multiple-retort stokers the region of air deficiency and smoke is found to be a narrow band very close to the front wall where the fuel bed is thickest. Repeated experience has shown that only enough penetration to supply air to this band is necessary. Probably the curves will apply to these cases if the penetration, L, is taken as the depth of this region, and the spacing is so adjusted that at least one jet discharges into the smoky region above each retort.

Similarly with underfeed stokers it is often noted that smoking is severe only over the retort. This region could likewise be traversed with jets selected for limited penetration.

Jets for Hand-Fired Furnaces

A certain hand-fired installation in an Eastern city has been operating successfully for a number of years. Its dimensions are as follows:

Boiler	250 hp
Pressure	120 psi gage
Firebox width	8.5 ft
Firebox depth	7.0 ft
Approximate burning rate at rating	18 lb per sq ft per hr

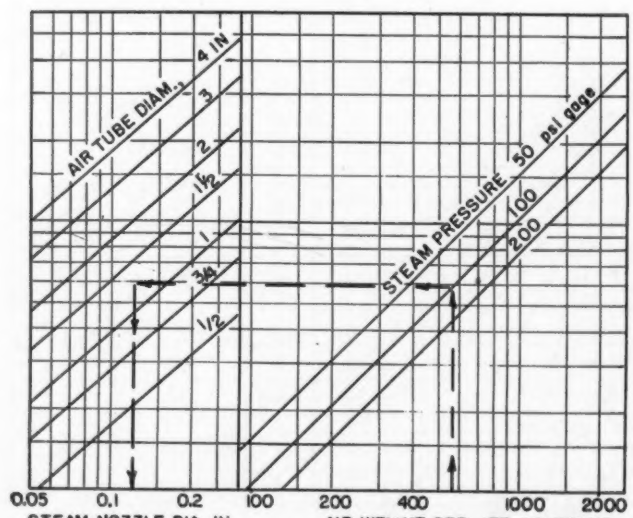

Fig. 3—Steam nozzle selection chart

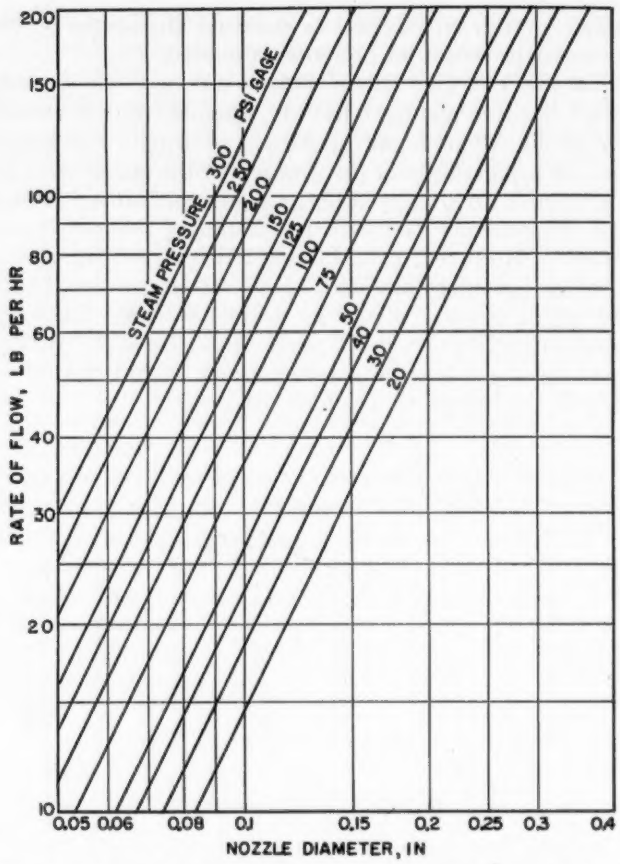

Fig. 4—Flow of saturated steam according to Grashof's Equation; atmospheric discharge

Number of jets	7
Air-tube diameter	1.8 in.
Steam nozzle	0.125 in.
Jet spacing	15 in.

According to Fig. 3 these jets each deliver 930 lb per hr, a total of 6510 lb per hr or approximately 60.6 per cent of theoretical air, 30.3 per cent of the total air when operating with 100 per cent excess. According to Fig. 2, this system would adequately supply a 12-ft grate with 37 per cent of theoretical air as overfire air at this rate. Table 1 shows the spacing should be 16 in.

This furnace might be said to be "overaired." However, the design of this installation before any engineering data were available will be held in higher regard when it is remembered that this is a hand-fired boiler and that, therefore, the effective burning rate fluctuates widely while the volatile matter is being distilled from the fresh coal. After each firing, the rate undoubtedly rises to several times the uniform rate of 18 lb per sq ft per hr. After this brief period, the jets are and should be turned off because overfire air is no longer needed. For the period during which the jets are used, this installation is probably not greatly "overaired" and, except for the apparently excessive penetration, is a good example of intelligent design. The excessive penetration has caused no difficulty, as the jets are above the fire door and pointed only slightly down to aim at the top of the bridgewall. The streams are deflected by the rising gases so that they pass harmlessly over the wall.

APPENDIX

The symbols used in this analysis are defined as follows:

W = weight of overfire air, lb per hr
R = burning rate, lb per sq ft per hr
r = percentage overfire air of theoretical required
A = theoretical air required, lb per lb coal
L = length of grate or depth to be penetrated, ft
S = spread of jet or width of section of firebox affected by one jet, ft (also spacing of jets, in.)
D = jet diameter or air tube diameter, in.
p = pressure of fan jets, in. water
T = temperature of overfire air, F abs
V = velocity, ft per min
E = entrainment ratio, lb of air per lb of steam
N = area ratio = $\dfrac{\text{area of air tube}}{\text{area of steam nozzle}}$
P = steam pressure, psi gage
a = steam nozzle area, sq in
d = steam nozzle diameter, in.

The amount of air, W, to be delivered over any section of width, S, and length, L, of a fuel bed is given by

$$W = R\frac{r}{100}ALS \tag{1}$$

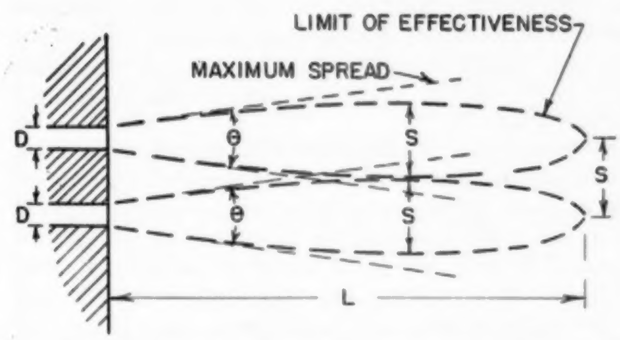

Fig. 5—Geometric relation between spread (or spacing), penetration and diameter of jet

S may be expressed in terms of the jet diameter, D and L, from consideration of the pattern of the jet. Fig. 5 shows the relation of spread to L and D. The angle θ is found to vary only slightly for various velocities[5] and can be considered constant for practical purposes. Hence:

$$S = D + \frac{L}{K} \tag{2}$$

The value of the constant K is found to be extremely variable, probably because the violently turbulent flow in the jet makes velocity readings very erratic. Values calculated are from dimensions scaled from the data of Zimm, Tegel and Cleve, reported by Cleve,[6] and from the data of the present investigation vary from 7 to 15. An average value of 10 seems to give spacings in rough agreement with practical experience, hence will be used until more precise values are available. If S and D are in inches and L in feet,

$$S = D + \frac{12L}{10} \tag{3}$$

Eliminating S, in feet, from equation (1) yields

$$W = R\frac{r}{100}AL\left(D + \frac{12L}{10}\right)\left(\frac{1}{12}\right) \tag{4}$$

This equation will yield, for a given installation, a value of D, inches, if W is known.

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[5] "Entrainment and Jet-Pump Action of Air Streams," G. L. Tube, G. B. Priester and D. K. Wright, Jr., *Heating, Piping and Air Conditioning*, vol. 13, pp. 708–715, 1941.
[6] "Die Wirkungsweise von Wirbelluftdusen," K. Cleve, *Feurungstechnik*, vol. 25, pp. 317–322, 1937.

From the previous work[1]

$$L\ (\text{inches}) = 0.51\,\frac{WT}{1000D}$$

or

$$L\ (\text{feet}) = 0.0425\,\frac{WT}{1000D} \tag{5}$$

combining equations (4) and (5) to eliminate W:

$$\frac{1000LD}{0.0425T} = R\frac{r}{100}AL\left(D + \frac{12L}{10}\right)\left(\frac{1}{12}\right)$$

or

$$D = \frac{1.2L}{\dfrac{28.24 \times 10^6}{RrAT}} \tag{6}$$

Equation (6) is the basis for Fig. 1.

The fan pressure to use for the specified conditions may be calculated from the usual relation between velocity and pressure for a carefully made, rounded approach nozzle:

$$V = 4006\sqrt{p} \tag{7}$$

for air at 70 F and 29.92 in. of mercury. From this

$$W = 98.35D^2\sqrt{p} \tag{8}$$

Combining equations (5) and (8) and solving for p:

$$p = \left(\frac{L}{2.217D}\right)^2 \tag{9}$$

Steam Jets

The entrainment ratio for a well-designed steam jet was found from the laboratory investigation to be given by

$$D = (0.89N^{0.56})0.953^{\frac{P + 14.7}{14.7}} \tag{10}$$

The weight of saturated steam flowing through any steam nozzle as given by Grashof's equation, lb per sec, is

$$W_s = \frac{a(P + 14.7)^{0.97}}{60.5} \tag{11}$$

This equation is the basis for Fig. 4.

From equations (10) and (11) the weight of air delivered is:

$$W = 41.85D^{1.12}d^{0.88}(P + 14.7)^{0.97}0.953^{\frac{P + 14.7}{14.7}} \tag{12}$$

This equation is the basis for Fig. 3.

Lamme Medal Award

The 1943 Lamme Medal of the American Institute of Electrical Engineers has been awarded to Arthur H. Kehoe, vice-president of the Consolidated Edison Company of New York, "for pioneer work in the development of alternating-current networks and associated apparatus for power distribution. It is expected that the medal and certificate will be presented to him at the summer meeting of the Institute in June at St. Louis, Mo.

Mr. Kehoe is a graduate of the University of Vermont, 1911 in electrical engineering, and began his career with the Consolidated Edison System Companies that year. In 1919 he was made superintendent of the transmission and distribution department of the United Electric Light and Power Company and two years later became electrical engineer of that company. He became vice-president of the United Company and the New York Edison Company in 1932 and has been in charge of construction and shops of the Consolidated Edison Company since 1936.

Soot-Blower Applications

The following notes are a brief review of current practice in removing ash and slag accumulations from the heat-absorbing surfaces of steam-generating units. They indicate the types of soot blowers adapted to use in different locations and comment on advancement in soot-blower design to meet present exacting conditions.

ALTHOUGH soot blowers have been used for many years, the last decade has seen greatly increased application as a necessary adjunct to modern steam-generating units. The number desirable in each case depends upon the size and design of unit, the operating conditions and the character of the fuel. Where formerly it was customary to employ only a few blowers, one now finds many units provided with a sufficient number so distributed as to reach practically all surfaces where dust, ash or slag may accumulate. Moreover, different locations call for different types of blowers.

This trend has accompanied the desirability of maintaining all heat-absorbing surfaces as uniformly clean as possible in order to maintain capacity, superheat and efficiency, and to minimize or avoid outage for manual removal of deposits.

It is not to be inferred, however, that blowing offers a complete solution to the problem. It is conceded to be effective in removing dry or spongy accumulations, but where the ash is sticky or slag has been formed by molten ash that has fused to the tube surface, hand lancing also becomes necessary in many cases. Hence, the settings of most large modern steam-generating units are amply provided with lancing doors and conveniently located operating platforms. Nevertheless, soot blowers have achieved a very definite rôle in present-day boiler operation.

In general, soot blowers are subjected to very severe service and as operating practice has become more and more exacting, soot-blower design has progressed accordingly. Advantage has been taken of advances in metallurgy through the employment of materials for blowing elements and bearings that will resist scaling, oxidation, distortion and growth under high temperatures. While heavy seamless steel tubing has proved satisfactory for temperatures up to 900 F, alloys of medium chromium content, calorized or chrome-clad elements become necessary where the temperatures may reach 1650 F; and above this special alloys of high chromium content are generally employed.

Not only is it necessary that the bearings of rotating elements be of suitable heat-resisting material but their attachment to the tubes must be such as to utilize fully the cooling effect of the tubes and thereby avoid distortion. Practice in this respect employs either welding the bearings to the tubes or specially machined clamps that will assure full metal-to-metal contact, in order to transmit the heat to the boiler tubes.

Several types of blowers have been developed and are in general use, each applicable to certain locations within the steam generating unit. The stationary or rotating element having jets spaced over its length finds application in the cleaning of convection surfaces where the gas temperatures are not too high, such as boiler tubes, economizers, air heaters and some superheaters. It is necessary that the nozzles be designed to suit the tube arrangement, otherwise most effective cleaning will not result. It is also important that the elements remain in proper alignment, despite movement of boiler tubes with respect to the walls, and to compensate for such movement, flexible connections are employed between the head and the blowing element. Misalignment may result in severe cutting or grooving of the tubes. Care should be taken not to direct the jets against baffles.

The blowing arc is determined by the surfaces to be reached and in some designs it is possible by means of cams to blow lightly or heavily over certain portions of the arc. On low-pressure installations full steam pressure is applied but where the drum pressure is high, a reducing valve or orifices may be employed to reduce the blowing pressure to suit conditions.

Retractable Type for High Temperature Zones

In zones of very high gas temperature, experience has shown maintenance to be excessive with the conventional fixed or rotating types of soot blowers, hence for such places these have given place to retractable blowers in which the blowing element is withdrawn from the gas path when not in use. These may have either a long or a short blowing element and are usually fitted with a small number of large nozzles to permit delivery of a large volume of the blowing medium at high velocity. The former are adapted to blowing surfaces not otherwise reachable or where the fixed types would be subjected to excessive punishment, as for instance roof tubes, the entrance to superheaters and slag screens. The latter, designated as mass or gun-type blowers, usually have one or two large nozzles at or near the end of the blowing element and are especially adapted for removing large localized slag accumulations. They are used to clean furnace walls, screen tubes and other radiant heat-absorbing surfaces. Slow advancement and rotation of the element may be manual, by means of a chain or worm, by means of an air piston and air motor or an electric motor with remote control. When not in use the nozzle is retracted within a notch in the furnace wall and in some cases provision is made to cool it by air. When employed to remove slag from water walls the nozzle projects only slightly into the furnace so that the jet is parallel to the wall tubes and thus peels off the accumulations.

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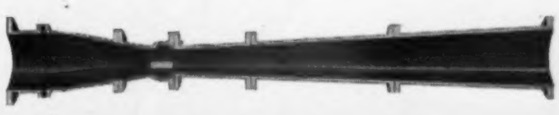

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BUILDERS-PROVIDENCE

Steam or air is ordinarily used as the blowing medium. Dry saturated steam is preferable to superheated steam, but wet steam that contains slugs of water is likely to produce cutting of metal surfaces reached. For this reason, also, care must be taken to see that the head and piping is properly drained before operating the blower.

Air is well adapted to marine boilers because of the desirability of conserving condensate. One type of blower, developed especially for marine use but also now applied to stationary units, delivers the air in intermittent puffs through successive nozzles and successive elements. The operation is entirely automatic, and is subject to selective remote control.

Mention may also be made of the telescopic type of retractable blower and those with special arrangements, such as pantagraphs, for reaching the surfaces of tubular air heaters.

An important feature of all soot blowers is the design of head and valve control which governs the admission of the blowing medium and rotates the element. They differ with the make, but it is not within the scope of the present article to describe their details.

Steam Blowing Pressures Vary

Practice varies widely as to steam pressures employed for blowing. A review of current practice among central stations shows this to range from 200 to 800 psi and in a few cases higher. Obviously, where the drum pressure is relatively low, the blowing pressure is limited. Quite a number of the high-pressure plants seem to find around 600 psi satisfactory for operating of the mass blowers and reduced pressures for cleaning other parts of the units. However, the optimum pressure seems to be a matter of trial for each plant and is affected by the design of unit, the operating cycle and the character of the ash deposits in different locations. Frequency of blowing also depends on local conditions. Gas temperatures, steam temperature and draft loss serve as a guide to the necessary frequency.

Although soot blowers have been greatly improved during recent years, it must be remembered that they are usually subjected to very severe conditions. Such being the case, they warrant frequent inspection and prompt attention when servicing is indicated, for their proper functioning is an important factor in the maintenance of high unit operating efficiency and reduced outage.

As previously mentioned, some types of slag do not respond readily to blowing and require hand lancing. For this purpose some companies have employed hot water, others compressed air, or a mixture of air and steam, for it is difficult to handle a hand lance with steam alone because of the temperature. Although water will most readily crack off slag accumulations, its use is potentially dangerous and in many instances has brought about disastrous conditions. Its impingement on hot metal will set up local strains, produce leaks in rolled joints, damage hangers and cause surface cracks which become progressive and point to ultimate failure. Moreover, it is difficult to supervise such lancing so that these dangers may be avoided. For this reason the use of water as a lancing medium has been discontinued by many and present well-considered opinion is that it should not be employed under any circumstances.

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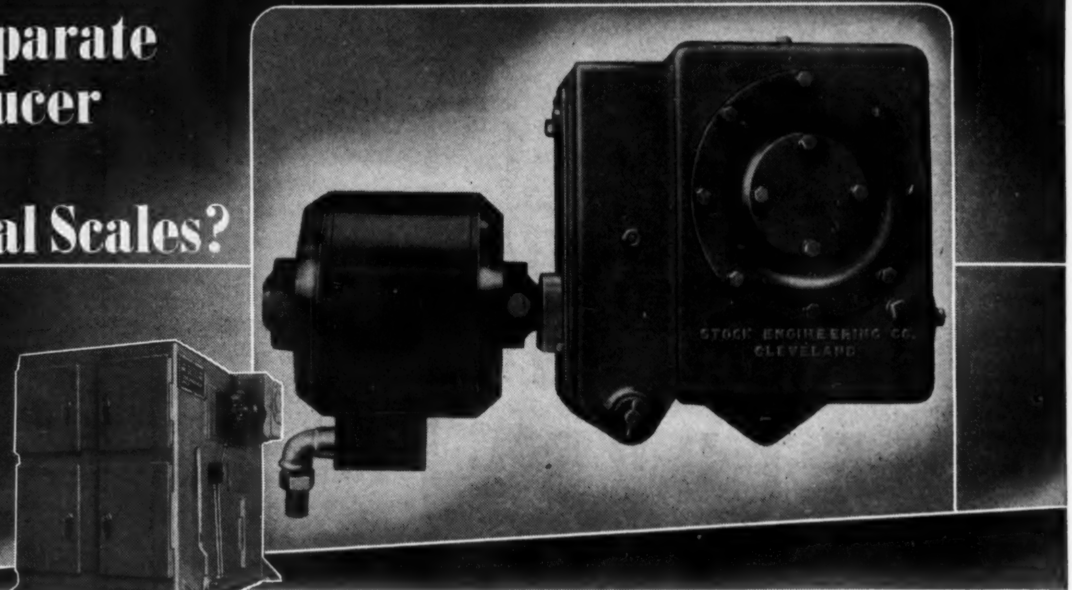

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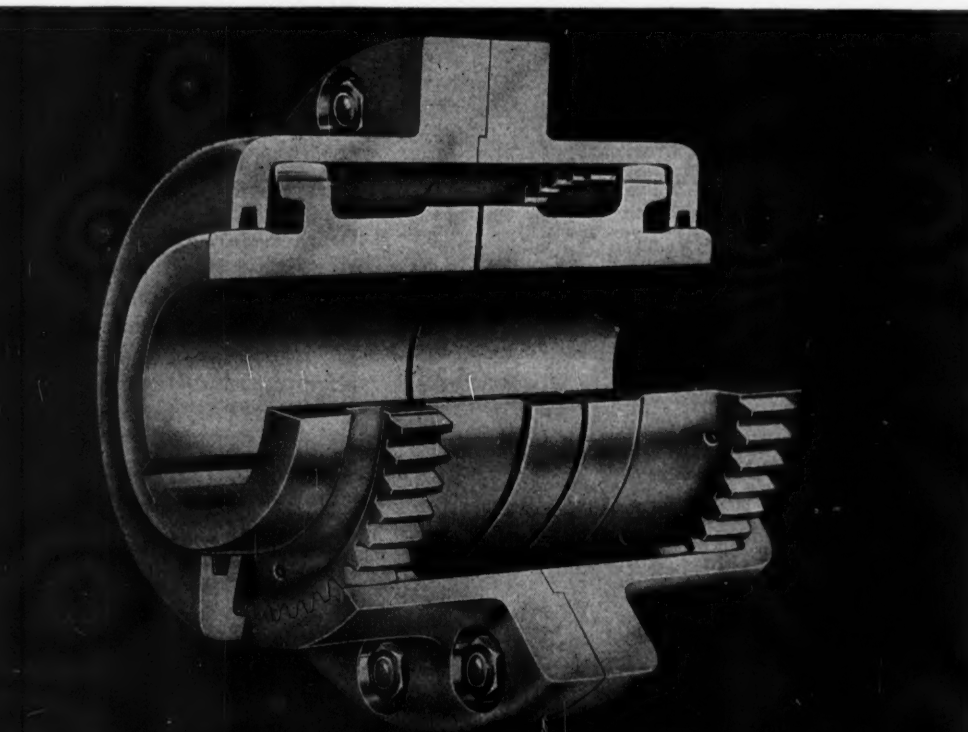

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The 1944 Coal Outlook

Speaking before the Chicago Coal Merchants Association on March 7, C. J. Potter, Deputy Solid Fuels Administrator, discussed the coal outlook for the present year. His remarks were, in substance and in part, as follows:

A careful examination shows that there has been a steady month-by-month increase in the rate of consumption, exclusive of seasonal variations, and it appears that in 1944 the all-time peak of bituminous coal requirements will be reached. In 1943 we actually consumed 610 million tons (part of which was taken from storage), but indications are that the coal-mining industry will not be able to produce more than 600 million tons during the present year. This indicates a probable deficit of 20 million tons which must be met by sound distribution and real conservation.

The best estimates of representatives of all producing fields are that, *if* the industry has no further manpower losses; *if* adequate supplies and equipment are available; *if* there are no work stoppages; *if* the normal effect of production can be overcome; *if* absenteeism among miners can be reduced; *if* there is no diminution of transportation services; *if* conservation is carried out; and *if* other intangible handicaps are not encountered, sufficient coal can be produced to get by.

Certain uses, such as the production of steel, coke and gas have rigid quality specifications which must be met in supplying coal, since they are vital to the war effort.

The stocking of coal by all users, including industrial and large commercial plants, is a program that must be vigorously pushed because it is essential to keeping all mines in full production throughout the warm months of the year. Since the mines have practically no facilities for the storage of bituminous coal, it will be necessary that the coal be moved currently as produced.

While there will be a general urge among most industrial consumers to store coal during the summer, there are signs that tonnages of some of the least desirable coals may go begging. However, if production requirements are to be met it will be necessary for industrials to store both the preferable and the less desirable coals. Where plants have access to coals that will not store, it is urged that they utilize these to the greatest possible extent for current consumption during the warm season and place the storage coals in their stockpiles.

With reference to anthracite, the deficit this winter of about five million tons of domestic sizes in the Middle Atlantic states and New England was attributable partly to a shift in population from rural to urban centers, with a resultant diminution in the cutting and use of fire wood; partly to the withdrawal of heavy tonnages of coke from domestic uses, and partly by conversions from oil to coal.

One of the measures taken to cope with this shortage was the prohibition during January, and curtailment during February and March, of anthracite shipments west of the Pennsylvania line. Another was the diversion of a limited amount of both high-volatile and low-volatile bituminous coals into the deficit areas. A limited flow of anthracite to consumers in the Middle West was permitted during most of the winter to take care of those whose equipment necessitated the use of this fuel.

On the assumption that the anthracite industry will not be able to shoulder much more of the war burden next winter than it did during the current winter, the Solid Fuels Administration is contemplating the issuance of regulations that would limit dealers' deliveries of anthracite to $87^1/_2$ per cent of the consumer's annual requirements, less the inventory which he had on hand April first. It is expected that this will also apply to the distribution of domestic coke.

In conclusion, Mr. Potter stated that it will be the policy of the Solid Fuels Administration to place conservation on a par with all other measures adopted to protect the fuel supply, and that preparations are now being made for launching a vigorous coal conservation program.

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Formation of Bonded Deposits on Economizer Tubes Explained

IT HAS long been observed that in many stoker-fired British power stations hard bonded deposits form on the gas-swept surfaces of economizers and air heaters, which are of such a character as to resist removal by soot blowing. Curiously, these deposits begin to appear only after the unit has been in service for six to twelve months and following this period of immunity they build up rapidly in a rough overlapping scale structure growing in the direction opposed to gas flow. Moreover, in none of the pulverized-coal-fired plants has this phenomenon been observed.

This condition was discussed and an explanation offered in a paper by J. R. Rylands and J. R. Jenkinson, presented before a joint meeting of the Institutions of Mechanical and Electrical Engineers in London on November 4, 1943, and subsequently reported in *Engineering* of December 24.

Three theories of deposits have previously existed; these are:

(1) The "Fused-Ash" theory which holds that the ash in the form of flying incandescent plastic particles impinge on the economizer surfaces and form a sticky base for the adherence of dust and ash and form a bond which solidifies into a hard bonded scale.

(2) The "Sodium Salt" theory, based on the fact that coal ash contains sodium salts which combine with sulphuric acid in the gases and form sodium sulphate. This condenses on the economizer tubes and with the dust and ash forms a hard bonded scale.

(3) The "Dewpoint" theory which considers that if the tube surface is below the dewpoint of the gases, moisture will condense and trap small dust particles and if the gases contain sufficient sulphur trioxide, the dewpoint may occur at higher temperatures.

In the opinion of the authors none of these theories explains the condition noted. If the fused-ash theory were to apply, the growth of deposits would be continuous and gradual from the time the unit went into service. Furthermore, such deposits would occur with pulverized-coal firing.

Most of the large number of economizer deposits examined showed them to contain only from one to two per cent sodium sulphate and the sodium-salt theory not only failed to explain the period of immunity but also did not account for the stalagmitic structure.

Under the dewpoint theory deposits would begin to form at the outset and, as the surface temperature rose with the thickening of layers of deposit, the rate of condensation of sulphur trioxide should diminish; hence the rate of accumulation should be less. This is not what happened. Neither does this theory explain the immunity of pulverized-coal-fired units.

Therefore, an entirely different explanation was offered by the authors. This was as follows:

During the period of immunity, the economizer tubes are gradually and directly attacked by the gases containing small amounts of sulphur trioxide and water vapor. Ferrous sulphate is formed, and this subsequently oxidizes to ferric sulphate. During this comparatively long period the tube surface appears to be clean, as the ferric sulphate is present only in minute quantities. Ferric sulphate, however, is not only deliquescent but is a powerful catalytic agent, capable of converting sulphur dioxide into sulphur trioxide. Thus, once a small deposit of ferric sulphate has been formed, the local production of more sulphuric acid from the relatively plentiful supply of sulphur dioxide and moisture can proceed at an accelerated rate.

There is, therefore, after a long period of what has been termed immunity, a thin film of sulphuric acid on the economizer tubes. This film collects a minute covering of dust and ash. If the acid concentration is sufficiently high, the dust may be attacked, aluminum sulphate formed, and the particles may be cemented together sufficiently to prevent their being dislodged by soot blowing. The question appears to be one of requisite acid concentration.

The strength of the sulphuric acid appears to be related directly to the tube temperature. It is known that sulphuric acid is formed by the catalytic process already mentioned, the constituents available being the sulphur dioxide and the water vapor in the gases. It is also known that the boiling point of sulphuric acid rises with the concentration, and that for each concentration there is a definite temperature at which the acid on the tube is in equilibrium with the sulphuric-acid vapor. Even an acid of 80 per cent concentration, by weight, has quite a low vapor pressure, and this is depressed still further in the presence on the tube of acid-soluble compounds. Therefore, at any given temperature there must be a definite acid concentration on the tubes.

Thus, at any given point on the economizer tubes where acid is being produced, its strength will depend on the temperature conditions prevailing on the metal or in the deposit. The specific quantity of acid at any particular place will depend on the rate at which the reactions are taking place and on the character of the gaseous constituents in contact with the actual metal. A deposit does not prevent the gases from making contact with the tube metal as examination has shown that even bonded deposits are porous to gases.

The authors point out further that no sulphuric acid can form on a surface much above 640 F, the approximate boiling point of pure sulphuric acid. Any dust collecting on such surfaces remains as dust and is practically unchanged throughout the run of the unit. Thus, in metal parts adjacent to the economizer tubes, the prerequisite condition for bonded-scale formation is absent and only dust will be collected and remain.

From the foregoing, the conclusion is reached that the dewpoint of the gases is only a minor consideration. It is not even a measure of the water vapor present and taking part in the reaction, since the dewpoint is very much modified by small quantities of sulphur trioxide.

Mercury Production and Price

The following, based on a recent announcement of the War Production Board, should interest those who have been following the fortunes of the mercury boiler.

The price of mercury skyrocketed when Germany invaded Poland, and in 1940 it sold on the New York market at close to $200 per flask. In 1942 OPA fixed a ceiling price of $197–199 per flask. But as supply increased, prices declined and reached as low as $130 in January 1944. This drop was precipitated by WPB's recommendation to the Metals Reserve Co. late in 1943 that it cancel its outstanding contracts with new mines or mines producing under 90 flasks in 1942.

However, it is felt by WPB that in order to safeguard stocks and to provide against contingencies, a segment of the domestic producing industry should be kept in substantial operation. Consequently, it recently conferred with producers at a meeting in Denver with the result that Conservation Order M-78 has been revoked and 4000 flasks per year of currently produced metal are to be allowed to serve hitherto prohibited uses.

At Denver it was estimated that at $150 per flask, domestic production could be maintained at about 46,000 flasks per year; at $130 the output would fall to 33,000 flasks and at $100 it would be 22,000 flasks per year. These figures represent price levels that would warrant certain mines to remain in operation.

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Laboratories

With a view to furnishing business organizations information on testing laboratories, the American Council of Commercial Laboratories has compiled a 36-page guide to leading independent testing, research and inspection laboratories with their geographical locations. A page is devoted to each, giving the names of officers and principal staff members, also outlining the facilities and the particular fields covered. A chart is included showing the services classified by commodities. Copies of the guide are obtainable from the Executive Secretary, 63 Wall Street, New York 5, N. Y.

Remote-Control Valves

The Grove Regulator Company has issued a 21-page catalog in two colors illustrating and describing its new "Flexo" remote-control valve which represents a distinct departure from the conventional types in that it employs the expansible-tube principle. The cylindrical flexive synthetic closure tube is hydraulically operated to shut off or open the flow through slots in a concentric core within the valve body. The whole valve takes up little more space than the pipe itself and is normally built for pressures up to 200 psi, although special valves of this type are furnished for very much higher pressures. The catalog also contains descriptions of the several models of Grove remote controls, regulators, reducing valves, stop-valves, etc., as well as dimensional tables.

Gages

Manning, Maxwell & Moore, Inc. announces a new 12-page bulletin descriptive of its phenol-turret case "Duragauge." It illustrates in colors the assembly of the gage and contains a full explanation in text of the rotary movement, the design and construction of the system, general dimensional drawings and illustrations of all thirty-one dials available.

Proportioners

A new catalog, No. 1100, has just been published by % Proportioneers, Inc. %, manufacturer of equipment for continuous automatic proportioning of fluids. This is devoted to a detailed description of "Adjust-o-Feeder" pumps and contains numerous diagrams in color as applied to typical boiler-water conditioning systems. Various accessories are also included as well as a capacity-schedule chart showing at a glance the maximum operating capacity in gallons per hour and discharge pressures of all "Adjust-o-Feeders."

Chain Belts

Bulletin No. 442 of the Chain Belt Company of Milwaukee is descriptive of a new conveyor-elevator system brought out by that company under the designation, "Rex Uni-Flo." This is a conveying unit of the continuous-stream type, composed of a chain belt having a closely spaced scraper-carrier flight, operating in an enclosed casing. The bulletin is fully illustrated by both diagrams and photographs.

Water Softening

Bulletin 30, issued by Cochrane Corp. is a reprint of an article by C. E. Joos, discussing improvements in the hot-process water softener. Covered are chemical reagents, reduction of natural sodium carbonate, removal of organic material and ammonia, applications and typical installations.

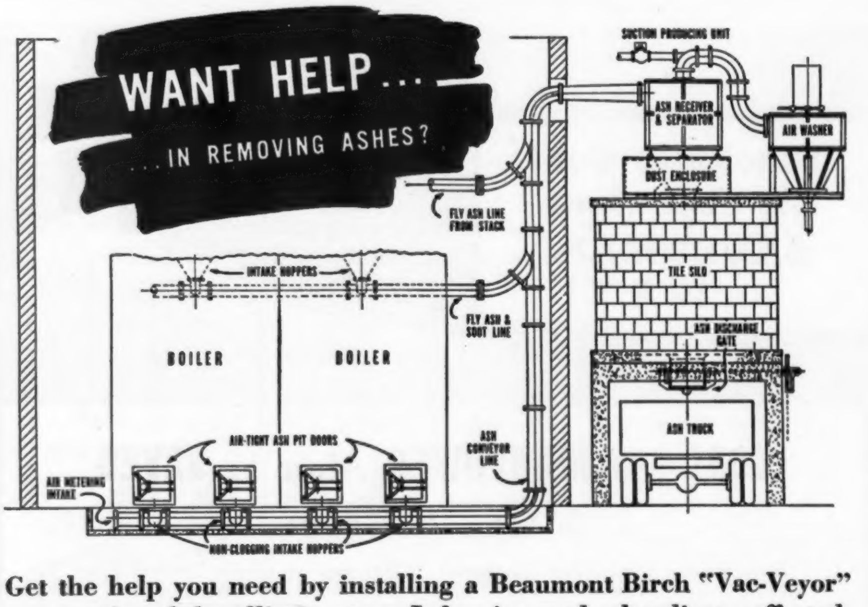
Zeolite Softeners

New 16-page Bulletin No. 603, published by Elgin Softener Corp., describes zeolite water softeners and other water-conditioning equipment. Included is a detailed description of newest type Elgin "Double Check" zeolite softener which it is said to give up to 44 per cent more capacity, prevent zeolite loss, cut backwash and rinse-water requirements and reduce salt consumption. It also tells how to increase capacity and efficiency of existing zeolite softeners. Other subjects covered are iron removal, filtration, aeration and boiler-water conditioning.

Soot Blowers

Vulcan Soot Blower Corp. has just put out an attractive 28-page catalog in color covering its complete line of soot blowers of various types. In addition to the descriptive matter, which is fully illustrated, typical application drawings are included showing these types as applied to different locations in numerous modern steam-generating units.

CORRECTION—On page 51 of the October issue of COMBUSTION the price of the book "The Steam Boiler Yearbook and Manual" was given as 20*s.* plus postage. We have since learned from London that the price of the 1943 edition is 30*s.* plus postage, or $6.50 approximately.



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A.S.M.E. Spring Meeting at Birmingham

April 3 to 5 is the date set for holding the Spring Meeting of the A.S.M.E. at Birmingham, Ala., with headquarters at the Hotel Tutwiler. While much of the program deals with aviation and war production, with special emphasis on the South, there will also be sessions on fuels, power and heat transfer. Following are some of the scheduled papers or talks that hold special interest for power engineers:

"A Study of Stoker Fuel Beds," by Otto de Lorenzi, Combustion Engineering Company. This will be illustrated by colored motion pictures.

"Federal Fuel Conservation Program," by T. C. Cheasley.

"Stress Distribution in Welded Joints as Shown by Transparent Models Under Polarized Light," by N. F. Bailey, of the Hartford Steam Boiler Inspection and Insurance Company.

"Panel Discussion" on Pulverizer Maintenance.

"Temperature Distribution within Boiler Tubing under Oblique Radiation," by Lt. Commander W. S. Kimball, U.S.N.

"The Influence of Through Metal on Heat Loss from Insulated Walls," by Victor Paschkis and M. P. Heisler, of Columbia University.

"Certain Aspects of High-Pressure Centrifugal Pumping Cycles," by Igor J. Karassik.

"Recent Axial-Flow Pumping Installations," by J. D. Scoville of S. Morgan Smith Company.

"Joint Operation of Steam and Hydroelectric Power Systems," by G. W. Spaulding, Pennsylvania Water & Power Company.

"Joint Operation," by A. T. Hutchins, Commonwealth & Southern Corporation, Birmingham.

There will be luncheon meetings on Monday and Tuesday at which talks will be given on subjects pertinent to the South, and R. M. Gates, President of the A.S.M.E., will be the speaker at the banquet on Tuesday evening.

Numerous plant inspection trips are being planned.

To Head Engineering Societies

Sewell H. Downs, Chief Engineer of Clarage Fan Company, has been elected president of the American Society of Heating and Ventilating Engineers.

C. A. Powel, Manager of the Headquarters Engineering Departments of Westinghouse Electric & Mfg. Company has been nominated for the office of president of the American Institute of Electrical Engineers.

Dr. Harry R. Ricardo, well known British consulting engineer and research worker, has been elected president of The Institution of Mechanical Engineers.

Books

1—The Marine Power Plant

By Lawrence B. Chapman

402 pages Price $4.00

In the second edition of this excellent text, the student will find a short but direct and thorough introduction to the fundamentals of the selection and design of steam and diesel propulsion plants. The marine engineer will find it refreshing with many helpful guides to design procedure and computation and to revealing comparisons for the proper selection of equipment. Owners and operators will find it similarly useful, and in addition may utilize it to check the results and correct the operation of their vessels.

Fundamentals and thermodynamic principles are set forth in a manner easily comprehended and descriptions of marine equipment are handled with clarity. In the selection of equipment, the latest practices are brought out but sight has not been lost of installations of the recent past—of ships that will be in operation for some years to come.

There are chapters on Fuels, Marine Boilers, Combustion, Reciprocating Steam Engines, Geared Turbines, Turbo-electric Drive, Diesel Engines, Comparisons of Types of Propelling Machinery, Condensers and Their Auxiliaries, Power Plant Layouts and Computations for the Power Plants of Merchant Ships.

2—Marine Pipe Covering

By W. W. Godwin

142 pages $5 \times 7^1/_4$ Price $2.00

This handbook, just off the press, is the first written for marine pipe coverers and for those who are learning the trade.

The book comprises eight chapters dealing with: Moulded Pipe Covering, Plastic Cements, Curved Block Covering, Flat Block Covering, Canvas Covering, Covering Boilers, Moulded Cork Covering, Hair Felt and Asbestos Rope. Many pages are illustrated. Paragraph headings in bold type make this book of 142 pages one of easy reference.

3—Plane Trigonometry Made Plain

By Albert B. Carson

389 pages $5^1/_2 \times 8^1/_4$ Price $2.75

This book deals with the fundamentals of plane trigonometry in greater detail than is done in most texts, and the discussions are presented in such a manner—accompanied by an unusually large number of figures and illustrative examples—that the student is enabled to comprehend how and why the principles are employed, and their practical application.

Beginning with definitions and principles of triangles, the subject is developed in eleven chapters to include inverse trigonometric functions and trigonometric equations. Most of the chapters conclude with pertinent questions and answers, a brief summary of the chapter and many practice problems. The remainder of the book is devoted to logarithmic and trigonometric tables and a very complete 5-page index.

4—Substitutes

By H. Bennett

Price $4.00

This book should be of interest to all branches of industry where shortages are an everyday problem. It describes new products, processes and substitute materials, and alternatives for the numerous raw materials which have become unavailable because of the war. The author is an expert in this field and has not only given of his wide knowledge of substitutes but has also included the results of the extensive experience of many chemists, engineers and technical workers.

The contents include: substitutes for metals, plastics, textiles, rubber, chemicals, drugs, resins, waxes, paints, oils, fats and many other products. A plan is given which shows how to determine whether a substitute is suitable or not, and whether it will stand up under conditions of marketing and use.

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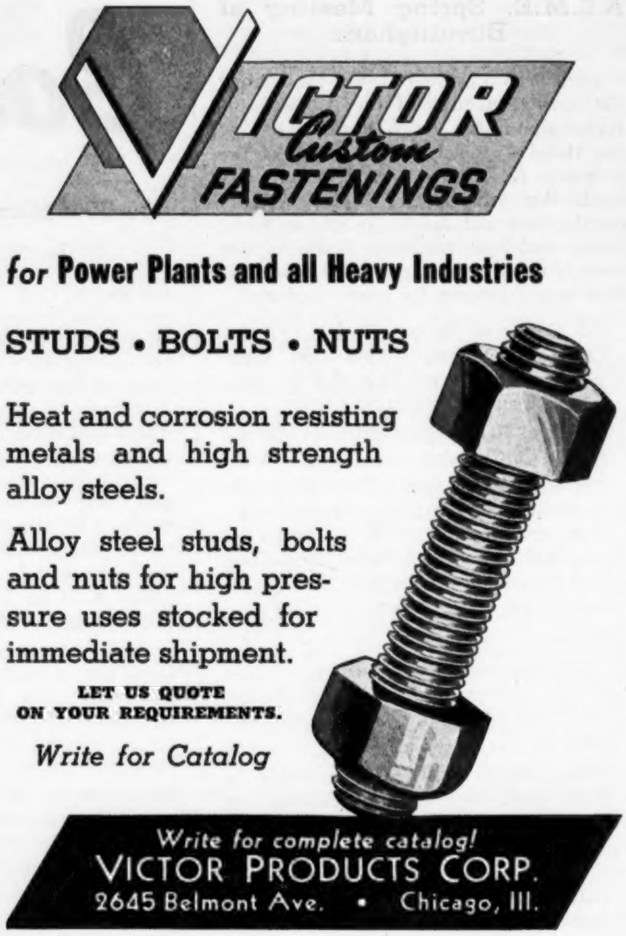

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Victor Products Corporation	60
Western Precipitation Corporation	Fourth Cover
L. J. Wing Mfg. Company	14
Yarnall-Waring Company	20